Session 1(b): Valves I

Session Chair

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Hopkinson Model 9054 Actuator Environmental Qualification and Testing

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Abstract

As part of Bruce Power's restart activities for Bruce Nuclear Generating Station "A", Units 3 and 4 - motor operated valves installed in our High Pressure Emergency Coolant Injection System required environmental qualification (EQ) upgrades, baseline maintenance and testing. The twelve inch Hopkinson parallel slide gate valves are operated with Hopkinson Model 9054 actuators. The actuator is controlled with limit switches only as the torque switch was removed from the control logic. This paper shares the results of the application calculations, EQ testing, actuator overhaul, actuator torque stand testing, and in situ differential pressure testing.

Introduction

This paper describes the steps Bruce Power had to take to qualify and return to service sixteen High Pressure Emergency Coolant Injection electric motor operated valves as part of our Bruce Nuclear Generating Station "A" Unit 3 and 4 Restart Project. This is an opportunity to share operating experience information on electric motor valve actuators that do not deal with Limitorque or Rotork with others in the Nuclear power industry.

Each operating Unit at Bruce Nuclear Generating Station "A" relies on eight Hopkinson Model 9054 electric motor operated valves to open allowing high pressure emergency coolant injection water to enter and cool the reactor. The valves are Hopkinson twelve inch, ANSI 900, NC1, parallel slide, venturi port gate valves. Bruce Power refers to these valves as D2O Isolation Valves as they isolate our heavy water Heat Transport System from the light water Emergency Coolant Injection System.

In 1993, the D2O Isolation Valves and actuators were modified to resolve reliability problems. The valve stem, yoke and anti rotation device were strengthened. The motor horsepower and output torque was reduced. The limit switch with torque switch back-up logic was changed to two out of three limit switch only logic (Torque switch was removed). One Limit switch was internal to the actuator and four are mounted on the yoke. These modifications allowed pullout torque to be available one hundred percent of the valve stroke and ensure the valves would survive the output torque and thrust.

Our Environmental Qualification Program had been suspended in 1997 due to Bruce A lay-up when Unit 3 and 4 were shut down and staff were reassigned within Ontario Power Generation. The EQ project had to be reactivated and completed as part of Bruce Power's Bruce A restart project. Bruce A's Hopkinson actuators were never previously environmentally qualified. Engineering had to choose between replacing the actuators or risking a test program to qualify them. Knowing that a Limitorque actuator could survive the test conditions even with its Nebula grease and its gaskets not needing to seal out the test environment, our Hopkinson actuator stood a good chance of success. We chose not to replace the actuators due to weak link concerns with the valve. We had just resolved them with the modifications mentioned above.

The Hopkinson representatives recommended some seal changes to protect the limit switch compartment and Hylomar sealant on joints. The motors would be rewound to the Bruce Power EQ specification. The limit switch would be replaced. A baseline overhaul would be completed. Due to resourcing conflicts, actuator overhauls were contracted out to the Hopkinson representative.

Findings:

Qualification testing –Actuator Steam environment, motor temperature test

A test actuator was subjected to a steam chamber at required accident temperature conditions (120 degrees Centigrade) and duration. The actuator performed its required safety function. The only casualty of the test was 2 of 8 micro switches used in the limit switch were wetted and failed. Our EQ engineering contractor decided it was easier to remove the internal limit switch from the poised logic circuit than to risk delays by iterative testing and correction. We would only use the internal limit switch for the test circuit to lower our exposure to pullout torque while performing tests.

Prior to the steam chamber test, we had rewound the motors to meet our EQ specifications. After the rewind, the motor was placed in an oven to bring its steady state temperature and subjected to a locked rotor torque test. A dynamic test was not possible in the rewind shop. No appreciable change in stall torque was noticed due to the elevated temperature.

Acceptance testing –Failures on torque test bench

All sixteen actuators were returned to the Station. The contractor completed internal inspections, replaced required bearings, upgraded the seals, and installed EQ motor and logic connections. They had even shipped a torque stand from England to test the actuators after they were rebuilt. The contractor was advised that we would be performing acceptance testing on our own torque test bench which allows us to measure actuator output torque with and without a thrust load applied. An allowable torque loss of less than ten percent of rated torque plus 1.4 foot-pounds of torque for every one thousand pounds of thrust applied is expected.

Bruce Power maintenance staff had experience on eight similar Hopkinson actuators previously tested and our torque loss acceptance criteria was achieved. With a thrust rating of 60,000 pounds, our loading criteria of using 54,000 pounds presented no apparent challenge to the actuators. This thrust rating was confirmed with Hopkinson many years prior and is included in many of their publications. Figure 1 shows Hopkinson's Actuator Division Data Sheet 70263 that confirms the rated thrust for a 9054 actuator.

The first actuator to be subjected to the torque stand testing was rejected immediately. While applying a compressive thrust load, the thrust bearing failed to carry the load. The drive shaft was being jacked right out of the actuator. A circlip had popped out of its retaining groove in the output shaft allowing unrestrained axial movement to occur. For this to occur so quickly under no load, it was suspected that the circlip was not seated in its groove allowing it to pop out. The circlip can be seen holding the sleeve in place on the output shaft above the helical wheel in the figure below. The circlip is required for the actuator to perform its open safety function.

The second actuator met the torque stand testing acceptance criteria.

The third actuator was able to complete unloaded thrust testing, but suddenly stopped rotating when the thrust bearing was loaded. The actuator had seized. Based on earlier experiences testing Hopkinson actuators, contact and galling between the thrust bearing and the output shaft were suspected. This is known to happen when the thrust bearing is installed incorrectly.

Testing the rest of the actuators continued in an attempt to obtain eight acceptable actuators to be used for our Unit 4. Only five of sixteen actuators ended up being accepted for service. Some were rejected for seized thrust bearings and some for having unacceptably high parasitic torque losses when thrust load was applied. Eleven bad actuators were prepared for return to the contractor for repairs. The contractor wanted all 16 returned, as they had no idea why some actuators were acceptable and others were not. The contractor was convinced we were overloading the actuator. We were convinced the contractor used non OEM parts to repair. All actuators were returned for re-inspection and repairs.

Circlip 23

The contractor disassembled all sixteen actuators. Sticking to the thrust overloading theory, they told us the actuators had a rated thrust of zero pounds and that we had overloaded circlip 23. This was an unbelievable statement coming from a manufacturer's representative who supplies rising stem gate valves and actuators! Circlip 23 (item 23 on actuator drawing) retains a sleeve with hammerblow lugs on it and is keyed to the output shaft. The sleeve and circlip also carry the tensile stem load on the thrust bearing in order to open a valve. The circlip had dished, indicating it had yielded. The contractor advised us that the only way the actuator would carry a thrust load was to replace the circlip with a split retaining ring or threaded collar modification. Our EQ contract engineers quickly sided with the manufacturer's representative. However, the thought of a modification did not appeal to us as this actuator had been in service for 20 years and we have 400 or more similar actuators in service. We also had documentation supporting our position that loading the actuator to 90% of rated thrust is not overloading it. Bruce Power told the manufacturer's representative contractor to recheck their calculations and verify the zero thrust comment.

Engineering investigation - Circlip application, shaft hardness, groove

Circlip 23 presented an engineering challenge- why did it work when Bruce Power's Maintenance department rebuilt and tested the actuators and fail when the contractor-repaired actuators were tested?

Bruce Power tested three output shafts and sleeves to see if we could yield a circlip in our maintenance shop. Our mechanics proceeded to load the sleeve, drive shaft and circlip to 61,655 pounds. The first test only caused the circlip to deflect 0.031 inch indicating the circlip was holding. Upon disassembly the circlip showed no signs of yielding only that shear contact had occurred. A second drive shaft only caused 0.028 inch deflection of the circlip when loaded. Again, no yielding was observed. A third drive sleeve finally revealed circlip bending – the clip was bending and sliding out of the retaining groove. The mechanics stopped applying load immediately.

Inspection of the sleeve revealed the edge contacting the circlip was not sharp. As a result, the circlip was experiencing a bending load instead of a shear load. The circlip groove in the drive shaft was also yielding. We measured the hardness of the drive shaft and estimated its yield strength to be near 65,000 pounds per square inch (psi).

We advised the contractor to inspect all the drive sleeve grooves and square up the sleeves to re-establish shear loading on the circlip and ensure the dimensions are within Hopkinson's allowable fits and tolerances. Skeptical that this would work, they agreed to try it and place an assembled output shaft, sleeve and circlip in their press, and press to thirty tons and proceed to the rated capacity of the press if the circlip held. They tested the assembly and were within manufacturer's allowable deflection. A load of ninety tons was applied and the circlip held although it did distort. The sleeve material yielded solid into the output shaft, which required machining to disassemble. The proof test was successful.

Based on the test results, Circlip 23 could once again be used for service. The circlip application was no longer in question. We had to purchase new output shafts and square up the sleeve surface or replace them to ensure the circlip was shear loaded.

Acceptance testing- ready for service

All sixteen actuators were overhauled and returned to Bruce Power. They were tested on our torque test bench. We disassembled any actuators that exceeded our parasitic loss criteria and improved bearing fits.

Typical pullout torque, stall torque and current readings at varying voltages are shown in Table 1. Our actuators were returned to the field acceptable for use.

Nuclear Safety Surprise – 5.5 MPa raised to 7.6 MPa DP Impact on Check Valve testing

The actuators have sufficient torque to open the D2O isolator valves based on our engineering calculations and uncertainties. Surprising results of a study performed by our Nuclear Safety Department concluded that some of the valves could see a higher differential pressure than originally expected due to the head pressure of our Heat Transport pumps. This raised the differential pressure from 5.5 Megapascals (MPa) (800 pounds per square inch differential (psid)) to 7.6 MPa (1103 psid) that four of the eight valves would be required to open against. This situation only becomes a risk if we depressurized a pipe section between the D2O isolators and a check valve in order to test stroke the check valve. Based on our extensive torque stand data, we were able to reevaluate our requirements. If the voltage was high enough, the actuators could still produce the required torque needed to open the valve. To confirm this, we had to determine our valve factor to ensure thrust capability was adequate by performing in situ differential pressure testing.

Our electrical engineers were able to determine that our voltage was high enough provided our class II inverters were available when the check valve testing was being conducted. This was added as a prerequisite to performing the check valve stroke test.

Differential pressure testing on four inlet header valves produced a 0.7 valve factor that we used for non differential pressure tested valve calculations. The high valve factor is higher than anticipated. Reasons for a high valve factor are:

- The D2O isolators have a nickel based hardfacing which Hopkinson calls "Platnam" instead of stellite.
- Differential pressure testing was done at a lower temperature and pressure than the valve would see at accident conditions.
- Instrumentation accuracy.
- Choice of mean seat diameter. The overlap of disc and seat was used to determine mean seat diameter.

Internal inspection history of these valves shows no signs of internal damage. The combination of actuator test data and differential pressure test data has been used to determine the valves will perform their safety function.

Conclusion

Through the use of qualification testing and the collection of actuator test data, Bruce Power was able to return all sixteen valves and actuators to nuclear safety service. The use of a torque test stand for electric motor operated actuators with controlled tensile and compressive thrust load capability located several operation problems. Most testing was done in a shop environment, minimizing the number of test strokes done at the valve. While the technical issues encountered are unique to Bruce Power's Hopkinson actuators, it demonstrates the work and knowledge provided by US utilities can be applied by others to improve equipment performance. The process allowed us to locate and neutralize a bad limit switch seal, reveal poor overhaul practices, resolve application problems, and collect test data to support safety analysis.

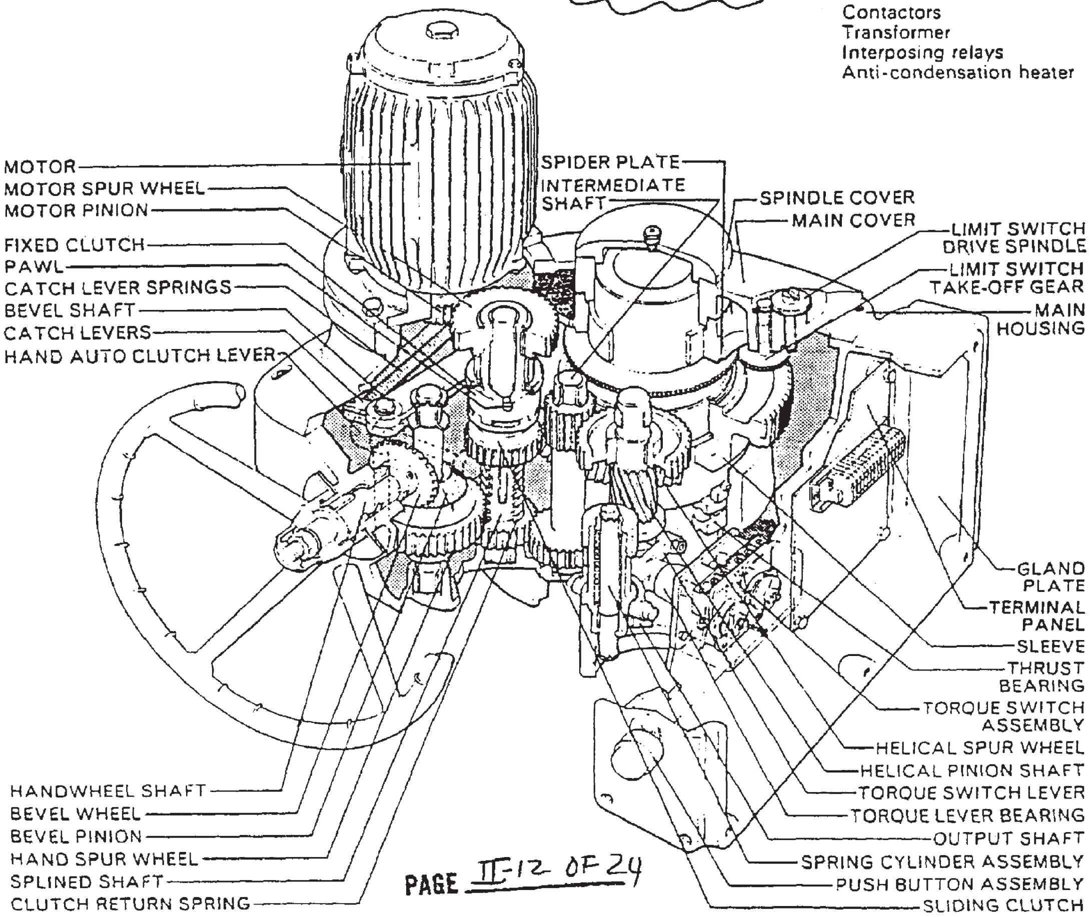
Figure 1

ELECTRIC ACTUATOR   FIGURE 9054

Standard Specification

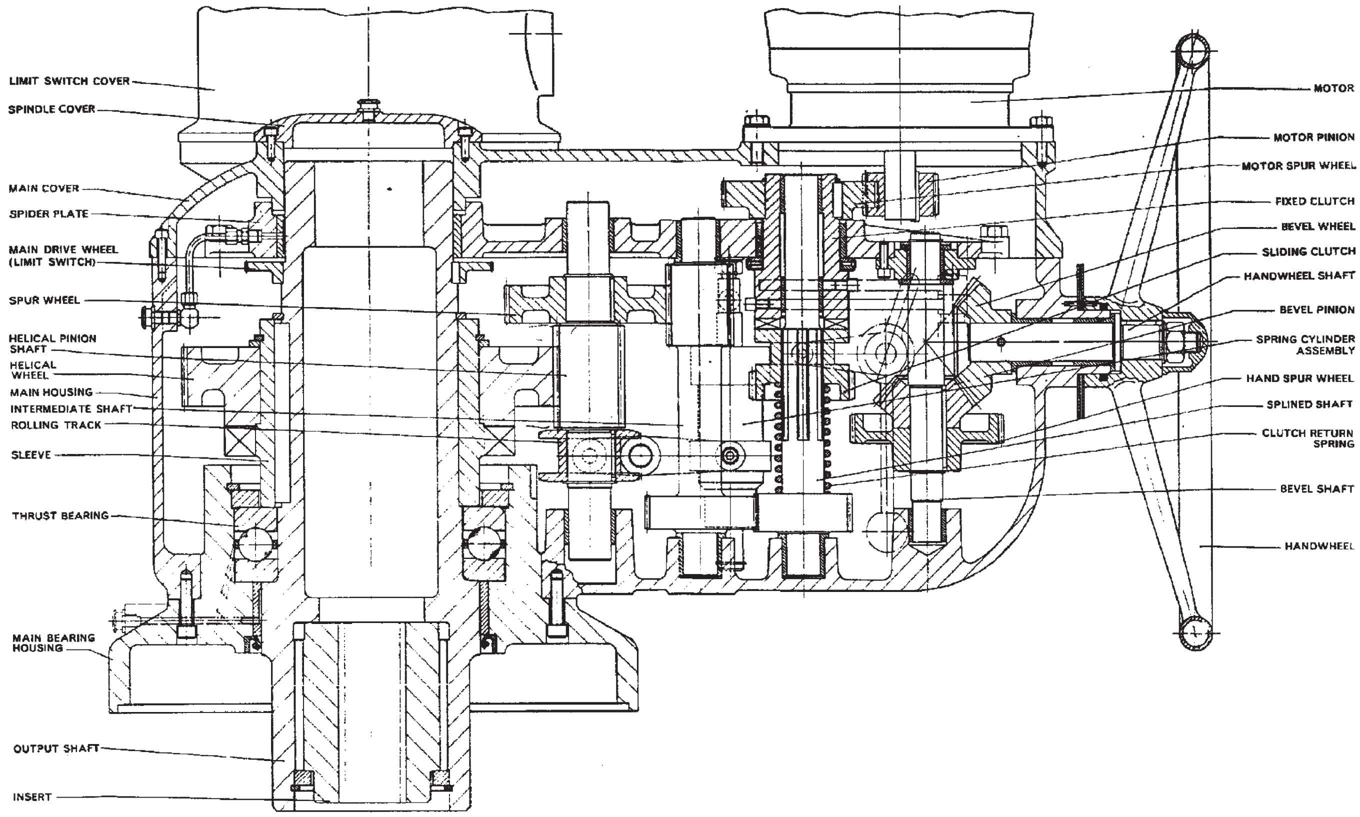
Output torque	600 lbf ft (814 Nm)
Output speed	24 rev/min (50 Hz) 29 rev/min (60 Hz)
Thrust	60,000 lbf (266 kN)
Maximum output shaft turns	100 Std. (1,000 special)
Drive	Detachable bronze or steel, external or internal sleeve
Maximum spindle (stem) acceptance	3" (76·2 mm)
Construction	Totally enclosed weatherproof to CSA enclosure 4, CEGB 569701 and IEC 144 (IP55)
Ambient temperature	70 °C maximum
Hand-wind ratio	10:1
Lubrication	Grease
Power supply	3 phase 50/60Hz
Motor	3 HP (2·2 kW) rating 30 minute valve duty. Speed 940 rev/min. * Insulation Class 'B'. Fitted with thermostat
Cable entries	Detachable undrilled gland plate
Weight	589 lb (267 kg)
Travel limit switches*	3 at Open position 2 at Close position
Torque limit switches*	1 in Opening direction 1 in Closing direction
*Single pole changeover type.	
Optional extras	Mechanical indicator Position transmitter Push buttons Isolator switch Selector switch Contactors Transformer Interposing relays Anti-condensation heater

* TYPICAL FULL LOAD 50HZ 6 POLE MOTOR



PAGE II-12 OF 24

Figure 2



SECTIONAL ARRANGEMENT OF GEAR BOX ASSEMBLY FOR FIG 9053/4 ACTUATOR

Table 1 Typical pullout torque, stall torque and current readings at varying voltages

Valve/ Voltage	Pullout torque in foot pounds/Amps rms	Stall torque in foot pounds/Amps rms	Parasitic torque loss in foot pound when thrust loaded
3-34330-MV6@ 591V	993/29.4	859/35.3	17
3-34330-MV6@ 565V	957/22.8	824/32.9	17
3-34330-MV6@ 450V	577/16	422/24.1	17
3-34330-MV6@ 400V	448	327	17

Entergy
Waterford 3 S.E.S
Hydraulic Operated Valve (HOV) Program

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Abstract

In general, Hydraulically Operated Valves (HOV) are the least populous of the Power Operated Valves at a Nuclear Power Plant. Motor Operated Valves (MOV), Air Operated Valves (AOV) and Solenoid Operated Valves are usually more numerous. Although small in population, HOVs are often used in important applications, especially when diverse modes of force are required. At Waterford 3 (W3), the six important HOVs are: Main Steam Isolation Valves (MSIV), Main Feedwater Isolation Valves (MFIV), and Shutdown Cooling Isolation Valves (SCIV). The MOV and AOV Programs have improved the reliability of MOVs and AOVs. A similar approach is being applied to HOVs. The three key elements of the HOV program are Design Basis Review, Diagnostic Testing, and Program Administration. Among these key elements, diagnostic testing of the HOV is the most difficult element. By applying knowledge from MOV and AOV testing, Waterford 3 has successfully implemented HOV diagnostic testing of selected valves. This program has been in place for the last two refueling outages. In the future, this testing may be extended to all six safety-related HOVs and also to Balance of Plant (BOP) valves. This presentation will focus on HOV diagnostic testing including the test method, test results, and resulting benefits that will improve HOV reliability and performance.

I. Background

In 2000, a number of Condition Reports (CRs) were issued to identify the problems associated with the SCIVs and MFIVs. Because of the above problems and considering the issues in NRC Regulatory Summary Issue 2000-03, "Resolution of Generic Issue 158: Performance of Safety Related Power Operated Valves under Design Basis Conditions," the W3 Business Plan assigned an action to Components Engineering to explore the feasibility of HOV diagnostic testing and the expansion of the AOV program to include HOVs. The intent of the action was to improve HOV reliability.

The feasibility study indicated:

Phase 1 – Design Basis Reviews (DBR):
Unlike the MOV and AOV Programs, the DBR calculations of all six safety related HOVs were previously approved.

Phase 2 – HOV Diagnostic Testing:
Prior to W3 RF 11 (April, 2002), Engineering studied the operation of safety related HOVs, combined testing techniques used within the MOV and AOV programs, and evaluated the available commercial diagnostic test systems. This study concluded that diagnostic testing of HOVs was feasible. During RF 11, HOV diagnostic testing began on the MFIVs and SCIVs.

Phase 3 – Program Administration: In progress.

II. HOV Diagnostic Test Equipment

In general, the testing techniques of MOVs are:

- Switch Actuation Monitoring: The actuation of torque switch and limit switches are monitored via current or voltage change.
- Motor Current Measurement: The motor current is monitored by a current (amp) probe.
- Motor torque is indirectly measured via the motor power or spring pack displacement which is correlated to a specific motor torque.
- Thrust/Torque Measurement: The stem thrust/torque is directly measured with permanently mounted strain gauge sensor on the stem. The stem thrust / torque could also be measured indirectly via a calibration file that is applied to the sensor readings (e.g., yoke mounted sensor, portable calibrator). The strain gauge is used to measure the valve stem thrust/torque.

The testing techniques of AOVs are:

- Pressure Measurement: The pressure sensors are used to measure the air pressure. In general, the maximum operating pressure of AOVs is approximately 120 pounds per square inch gage (psig).
- Thrust Measurement: The same strain gauge technique of MOVs is used on AOVs.
- Travel Transducer is used to measure the stem position during travel.
- In addition to the above, current probe, voltage measurement, Gauss sensor and acoustic sensor can also be used to monitor the Solenoid Operated Valve (SOV) operation and/or desired signals.

Criteria for Selecting HOV Diagnostic Test System/ Components

The components of HOV actuators are accumulators, SOVs, pneumatic valves, air or electrical pumps, pilot hydraulic valves and their control logic circuits. As a result, the HOV diagnostic test system requires the combined techniques of AOVs and MOVs. The HOV diagnostic equipment should have the following capabilities:

- High pressure measurement (hydraulic and nitrogen): the diagnostic system and pressure sensors shall be capable of acquiring high pressure data. The HOV pressure could exceed 5,000 psig.
- High thrust measurement: The output thrust of an HOV is much higher than the output thrust of an AOV or MOV. The HOV thrust could easily exceed 100,000 lbs.
- The measurement data are obtained and displayed in the same time reference.
- All other sensor measurements of AOV and MOV test equipment (e.g. travel transducer, current probe and voltage sensing device, Gauss sensor and acoustic sensor).

III. Shut Down Cooling Isolation Valves

Waterford has two SCIVs with one valve for each train. Each valve is located inside containment and between the Reactor Coolant System (RCS) isolation valves and outside containment isolation valves (SI 401A/B and SI 407A/B). This valve has an active safety function to close and remain in the close position during a Containment Isolation Actuation Signal (CIAS). This valve also has safety function to open fully and remain open under post accident Shut Down Cooling (SDC) entry conditions at 200F containment temperature. The open function is interlocked with pressurizer pressure to prevent over pressurization of the Low Pressure Safety Injection (LPSI) piping. The valve and actuator are designed as follows:

SCIV Size/Type	Design Pressure Unit: Pound per square inch gage (psig)	Design Temp	**Design Closing Thrust**
14” Flex Wedge Gate	2485 psig	650°F	33,819 lbs (Ref: Waterford ECM91-076 Rev 2)
Actuator	Normal Position	Failure Position	**Hydraulic Pump Max Operating Pressure**
Paul Munroe	Locked Closed	Closed	3000 psig

Description of SCIV Actuator

The valve is opened by the hydraulic force that acts on the bottom side of the piston. The valve is closed by the nitrogen pressure acting on the top side of the piston providing a store motive force. Upon initiation of a closed signal, four trip SOVs relieve the hydraulic pressure under the piston and drain the hydraulic fluid back to the reservoir.

Results & Benefits of SCIV Diagnostic Testing

Testing Results:

- Quickly identified problem (e.g., pump capability, internal leakage)
- Obtained dynamic response of nitrogen and hydraulic pressure
- Verified pressure switch settings
- Confirmed proper operation of sub-components (SOV, pneumatic valves etc.)

Benefits:

- Effective tool for future trending of hydraulic pump and SOV performance or for detecting other degradation (e.g., seal leakage)
- Condition monitoring in lieu of time based preventive maintenance
- Confirmation of sub-component operation helps eliminate and minimize Preventive Maintenance (PM) tasks

IV. Main FeedWater Isolation Valves

Waterford has two Main Feedwater Isolation Valves (MFIV), one for each redundant train. This valve has an active function to close under Feedwater or Main Steam Line Break (FWLB / MSLB). The valve requires a five-second closure per Technical Specifications.

The valve and actuator are designed as follows:

MFIV Size/ Type	Design Pressure	Design Temp	Stem Diameter
20" Double Disc Gate	1400 psig	480°F	3.75 inches
Actuator	Normal Position	Failure Position	Design Closing Thrust w/ Two Accumulators
Hydraulic/ Pneumatic (Anchor/ Darling)	Opened	Fail "As Is"	108,525 lbs

Description of MFIV Actuator

The MFIVs are controlled by hydraulic actuators. These actuators utilize a hydraulic/pneumatic control system with accumulators in conjunction with 3 way SOVs and 4 way hydraulic (pilot) valves to control hydraulic pressure within the actuator and thus open and close the valves. The valve accumulators (2) are precharged with nitrogen and then hydraulic fluid is added to achieve the desired operating pressure. Eleven gallon accumulators with integral piston stop tubes have been installed to provide a controlled volume in which to measure the nitrogen pressure. Both accumulators are required to actuate during FWLB/MSLB conditions for rapid valve closure. The 4 way hydraulic valves which control the flow path of hydraulic fluid within the actuator assembly are air operated. Solenoid operated valves control the air to the 4 way hydraulic valves, to direct hydraulic fluid flow. The MFIV are designed to "Fail As Is" on loss of electrical or air supply. Therefore, air accumulators are installed to ensure valve closure after a loss of instrument air. These accumulators are to ensure the valves can be closed within 1.5 hours from accident initiation.

Results & Benefits of MFIV Diagnostic Tests

Testing results:

- The initial diagnostic test revealed that after MFIV successfully closed, there was no closing force to maintain the valve in the close position. This behavior was similar to an MOV actuator with a non-locking gear set.
- The measured closing force with two accumulators (~ 110,000 lbs) agreed with the design closing force of 108,525 lbs.
- The bottom piston hydraulic pressure was significantly lower than expected for the MFIV.
- Confirmation of sub-component operation helps eliminate and minimize PM tasks.

Deficiency Identification:

- Non-locking closure stem force was corrected by modification.

Other benefits are:

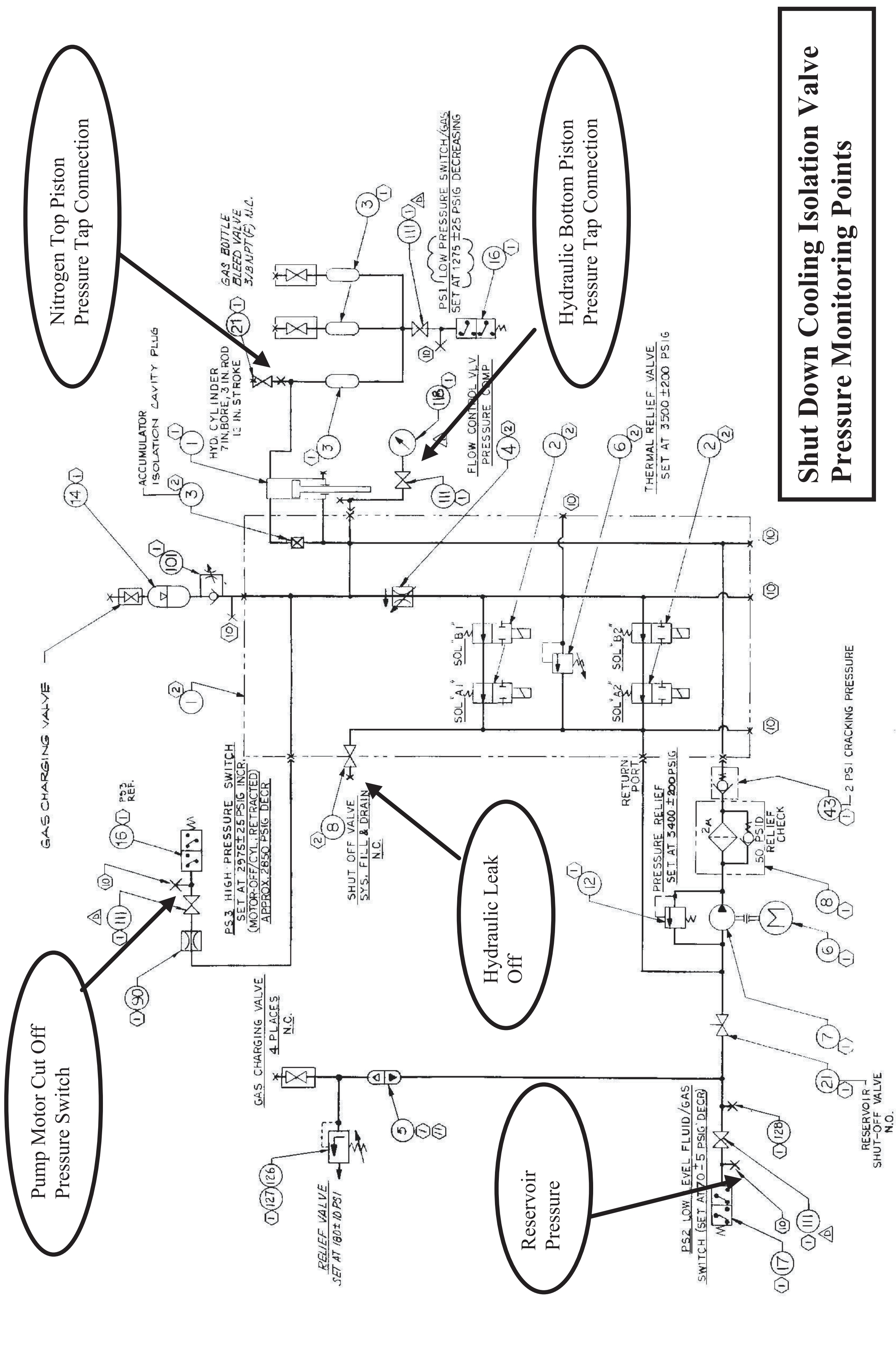
- Effective tool for future trending of the control pilot valves (SOV & pneumatic valves).
- Effective tool for future trending of other degradation (e.g., leakage).

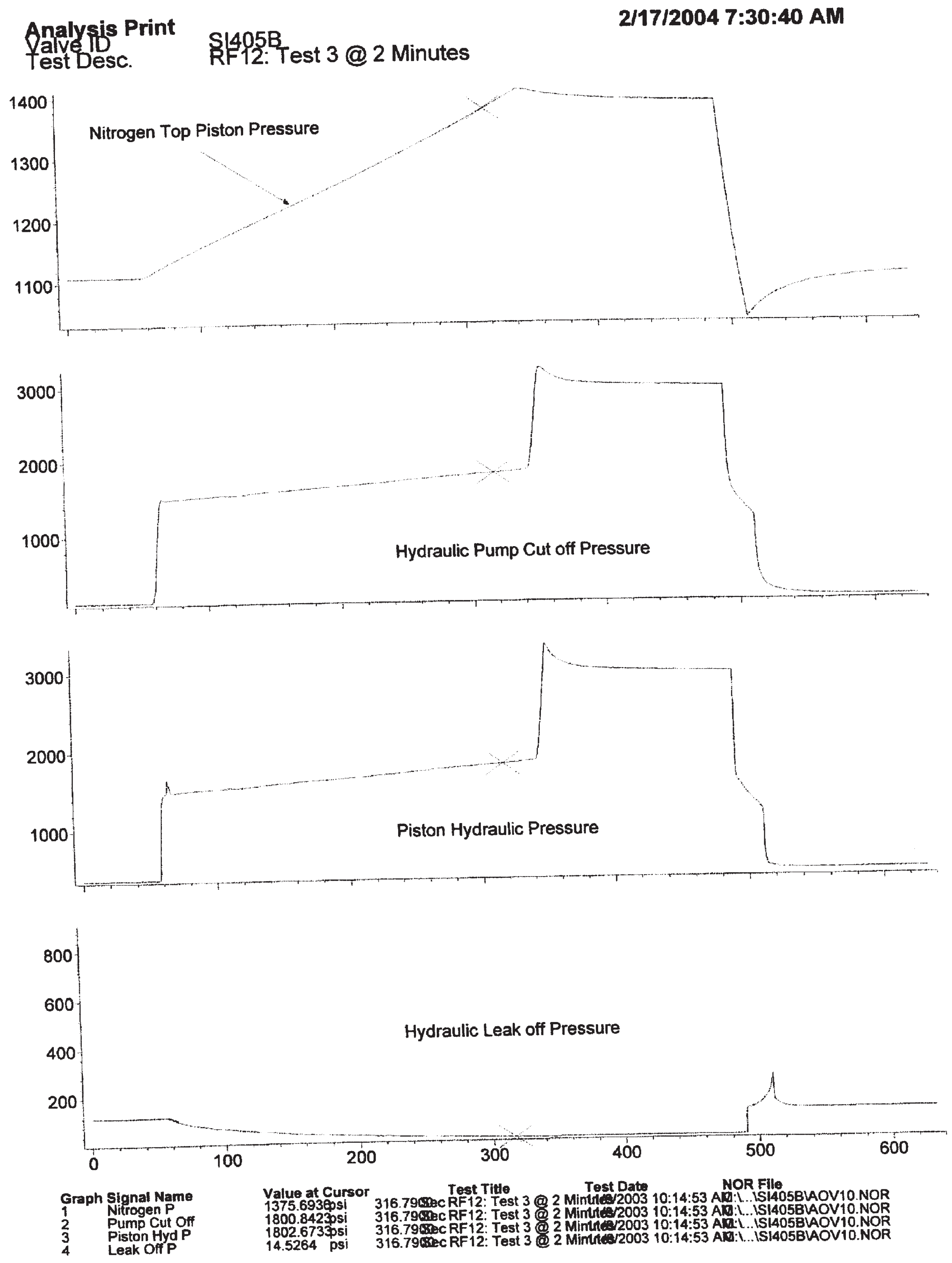
FUTURE ACTIVITIES

1. Perform HOV diagnostic tests on Main Steam Isolation Valves.
2. Apply HOV testing method to Balance of Plant (BOP) valves (e.g., main turbine isolation / throttle valves, Moisture Separator Reheater (MSR) intercept valves).

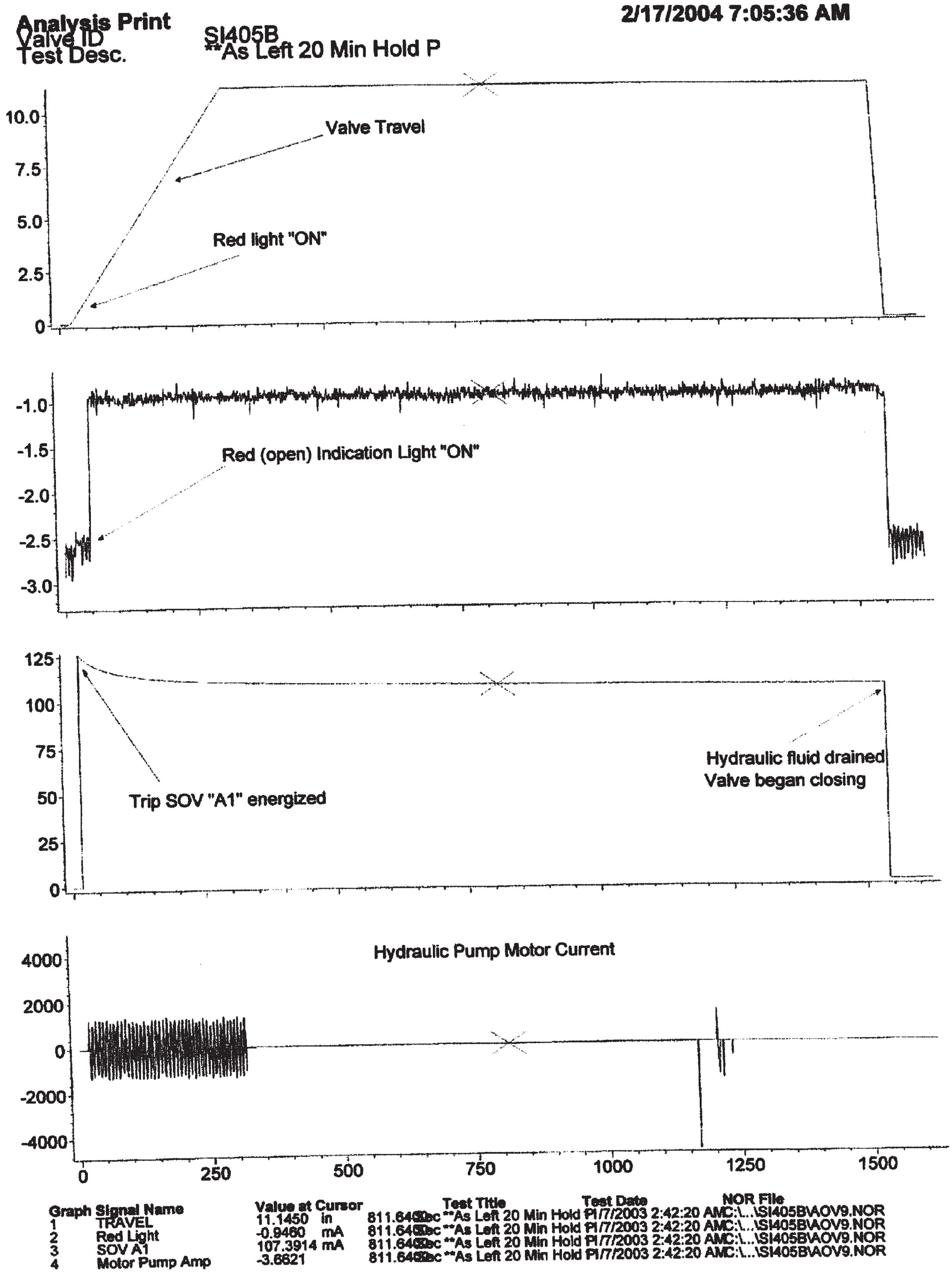
V. Conclusions

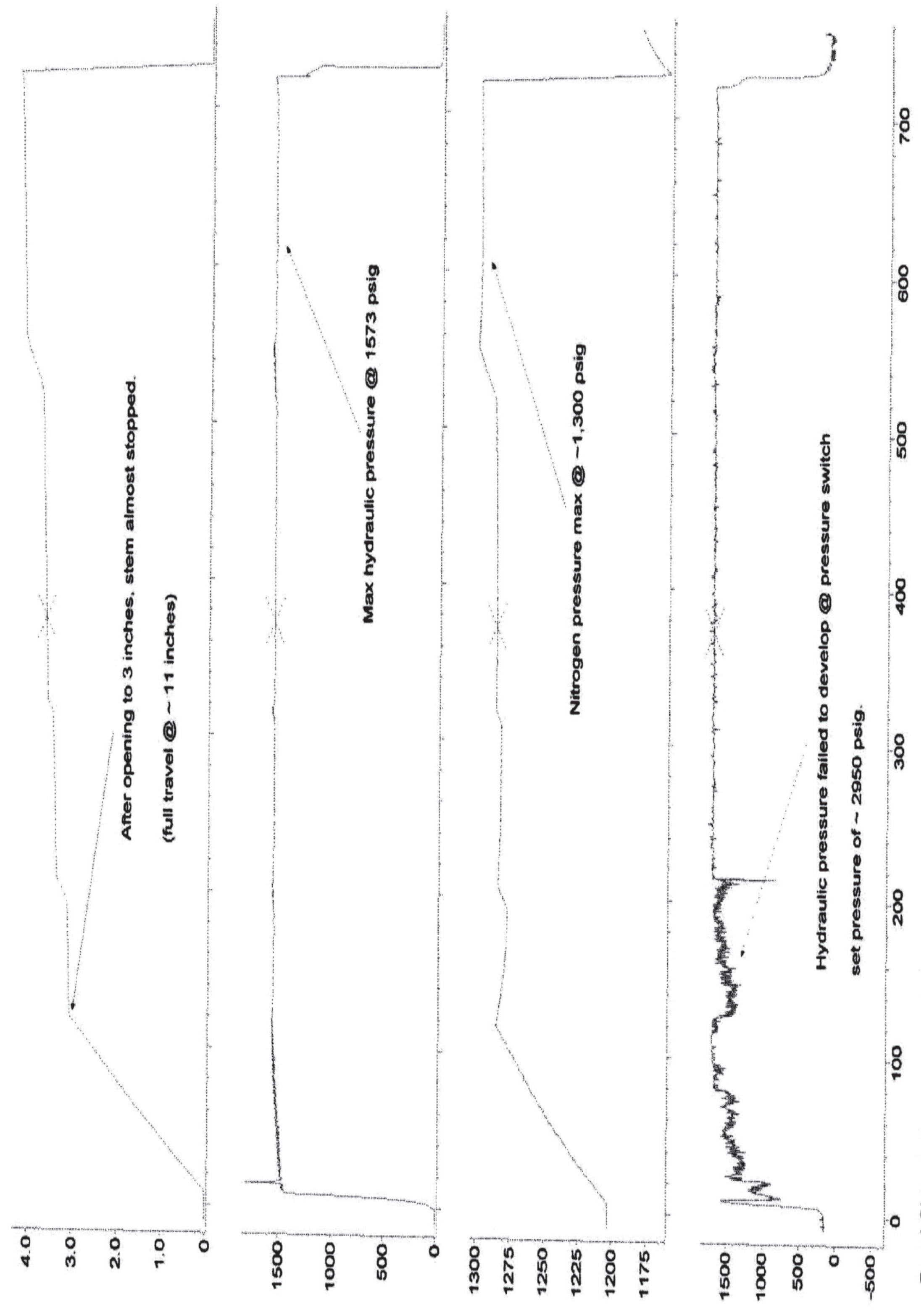
1. The benefits of MOV / AOV diagnostic testing are applicable to HOVs. HOV diagnostic testing is an excellent tools for:
   - ❋ Troubleshooting
   - ❋ Trending
   - ❋ Verifying HOV settings
   - ❋ Evaluating actuator output thrusts
2. Utilizing the HOV diagnostic testing should improve HOV reliability in the same way as MOV & AOV programs.
3. Because of high hydraulic / gas pressure and stem force, HOV diagnostic testing shall require extra cautions / attention.


Shut Down Cooling Isolation Valve Pressure Monitoring Points
Nitrogen Top Piston Pressure Tap Connection
Hydraulic Bottom Piston Pressure Tap Connection
Pump Motor Cut Off Pressure Switch
Hydraulic Leak Off
Reservoir Pressure
GAS CHARGING VALVE
ACCUMULATOR ISOLATION CAVITY PLUG
HYD. CYLINDER 7 IN. BORE, 3 IN. ROD 1½ IN. STROKE
GAS BOTTLE BLEED VALVE 3/8 NPT(F) N.C.
PS1 LOW PRESSURE SWITCH/GAS SET AT 1275 ±25 PSIG DECREASING
FLOW CONTROL VLV PRESSURE COMP
THERMAL RELIEF VALVE SET AT 3500 ±200 PSIG
SOL "A1"
SOL "B1"
SOL "A2"
SOL "B2"
PS3 HIGH PRESSURE SWITCH SET AT 2975±25 PSIG INCR. (MOTOR-OFF/CYL. RETRACTED) APPROX. 2850 PSIG DECR
PS3 REF.
SHUT OFF VALVE SYS. FILL & DRAIN N.C.
RETURN PORT
PRESSURE RELIEF SET AT 3400 ±200 PSIG
50 PSID RELIEF CHECK
2 PSI CRACKING PRESSURE
GAS CHARGING VALVE 4 PLACES N.C.
RELIEF VALVE SET AT 180± 10 PSI
PS2 LOW LEVEL FLUID/GAS SWITCH (SET AT 70 ±5 PSIG DECR)
RESERVOIR SHUT-OFF VALVE N.O.

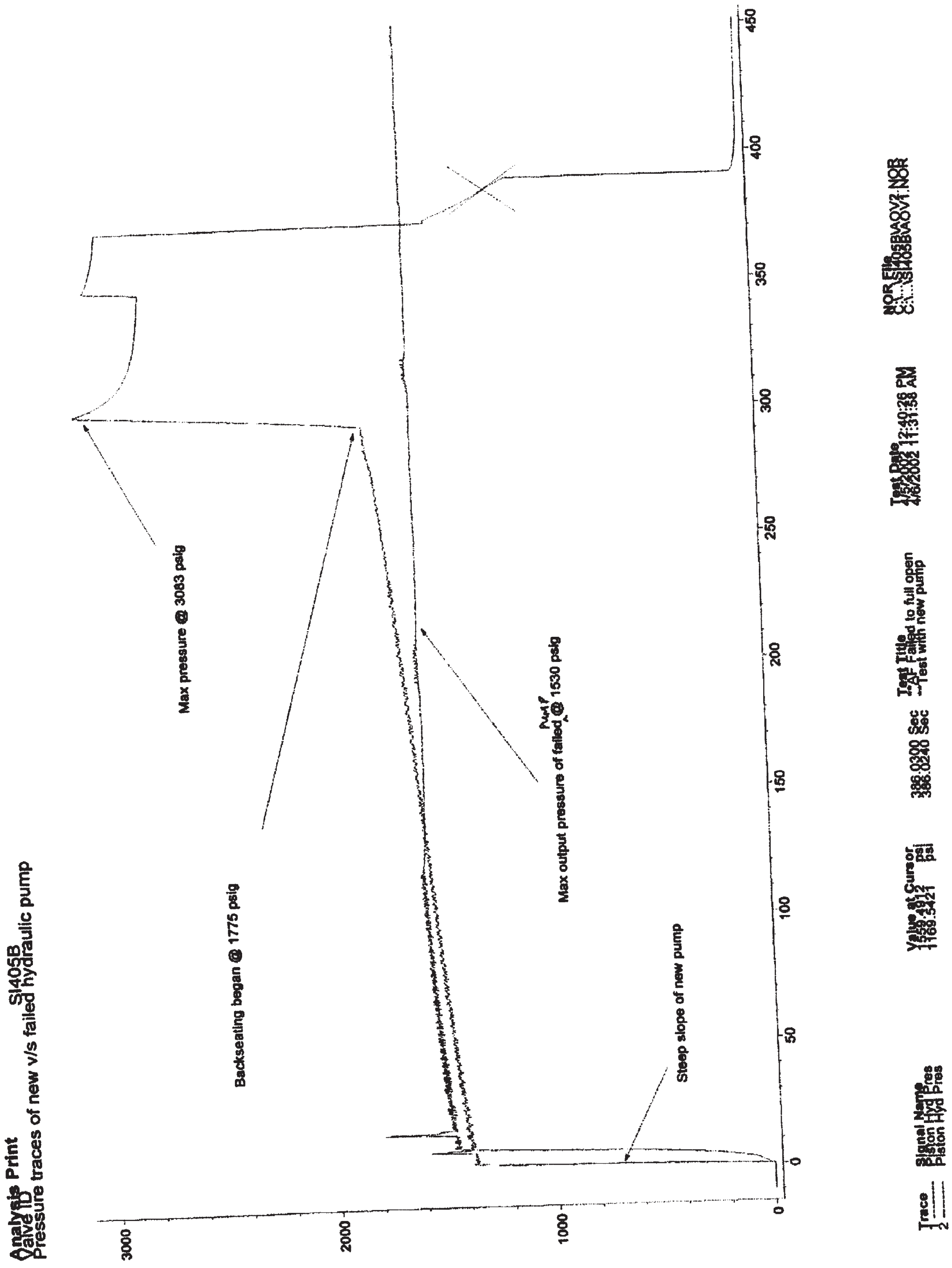


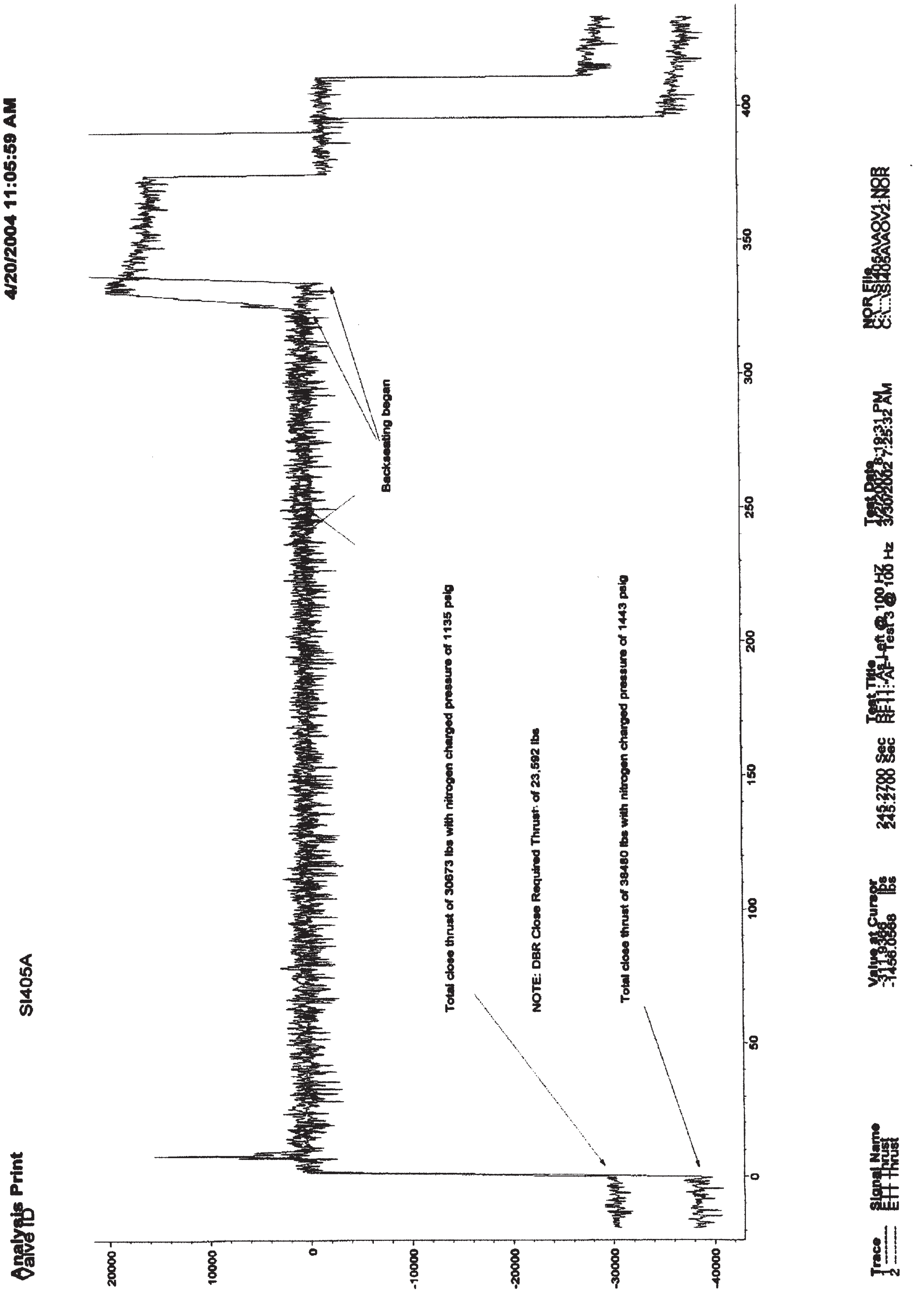
Shut Down Cooling "B" Valve


Analysis Print
2/17/2004 7:05:36 AM
Valve ID SI405B
Test Desc. **As Left 20 Min Hold P
Valve Travel
Red light "ON"
Red (open) Indication Light "ON"
Trip SOV "A1" energized
Hydraulic fluid drained
Valve began closing
Hydraulic Pump Motor Current
Graph Signal Name
1 TRAVEL
2 Red Light
3 SOV A1
4 Motor Pump Amp
Value at Cursor
11.1450 in
-0.9460 mA
107.3914 mA
-3.6621
Test Title
Test Date
NOR File
**As Left 20 Min Hold P
1/7/2003 2:42:20 AM
C:\...\SI405B\AOV9.NOR

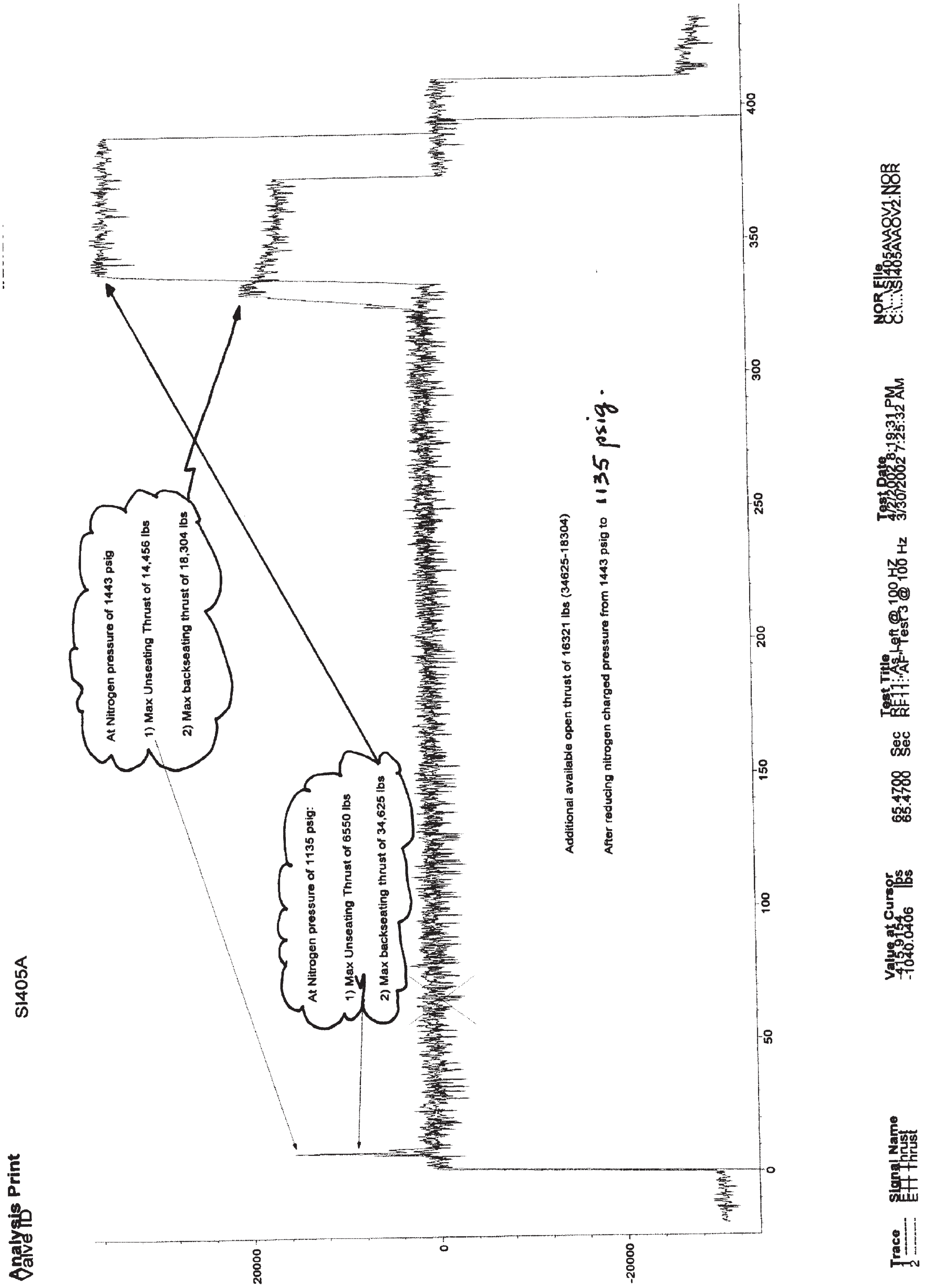


Shut Down Cooling "B" Valve Hydraulic Pump Failure

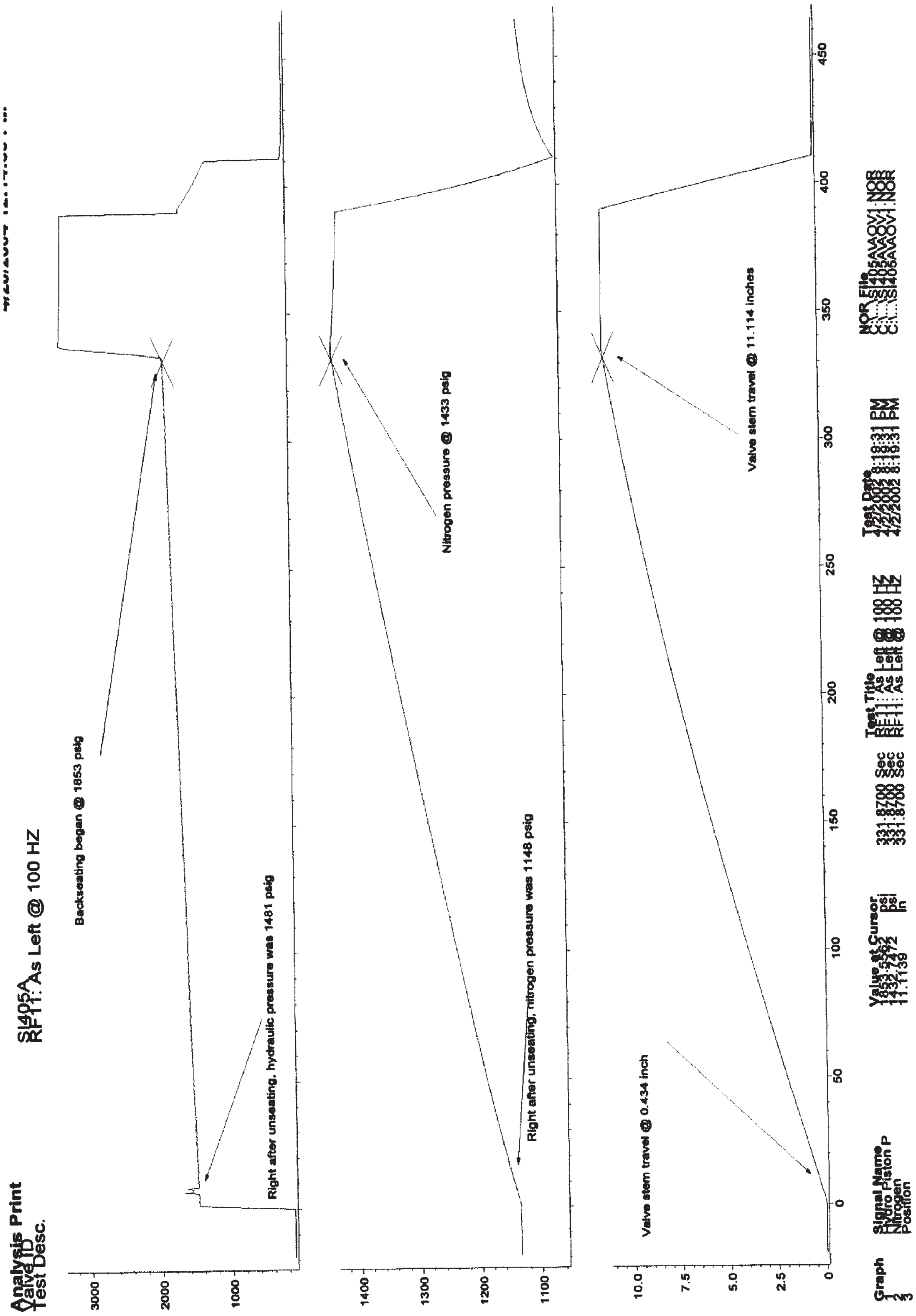

Analysis Print
Valve ID SI405B
Pressure traces of new v/s failed hydraulic pump
Backseating began @ 1775 psig
Max pressure @ 3083 psig
Max output pressure of failed pump @ 1530 psig
Steep slope of new pump
3000
2000
1000
0
0
50
100
150
200
250
300
350
400
450
Trace Signal Name Value at Cursor Test Title Test Date NOR File
1 Piston Hyd Pres 1529.4912 psi 386.0300 Sec ~AF Failed to full open 1/5/2002 12:40:26 PM C:\...\SI405B\AOV2.NOR
2 Piston Hyd Pres 1769.5421 psi 386.0240 Sec ~Test with new pump 4/6/2002 11:31:58 AM C:\...\SI405B\AOV1.NOR

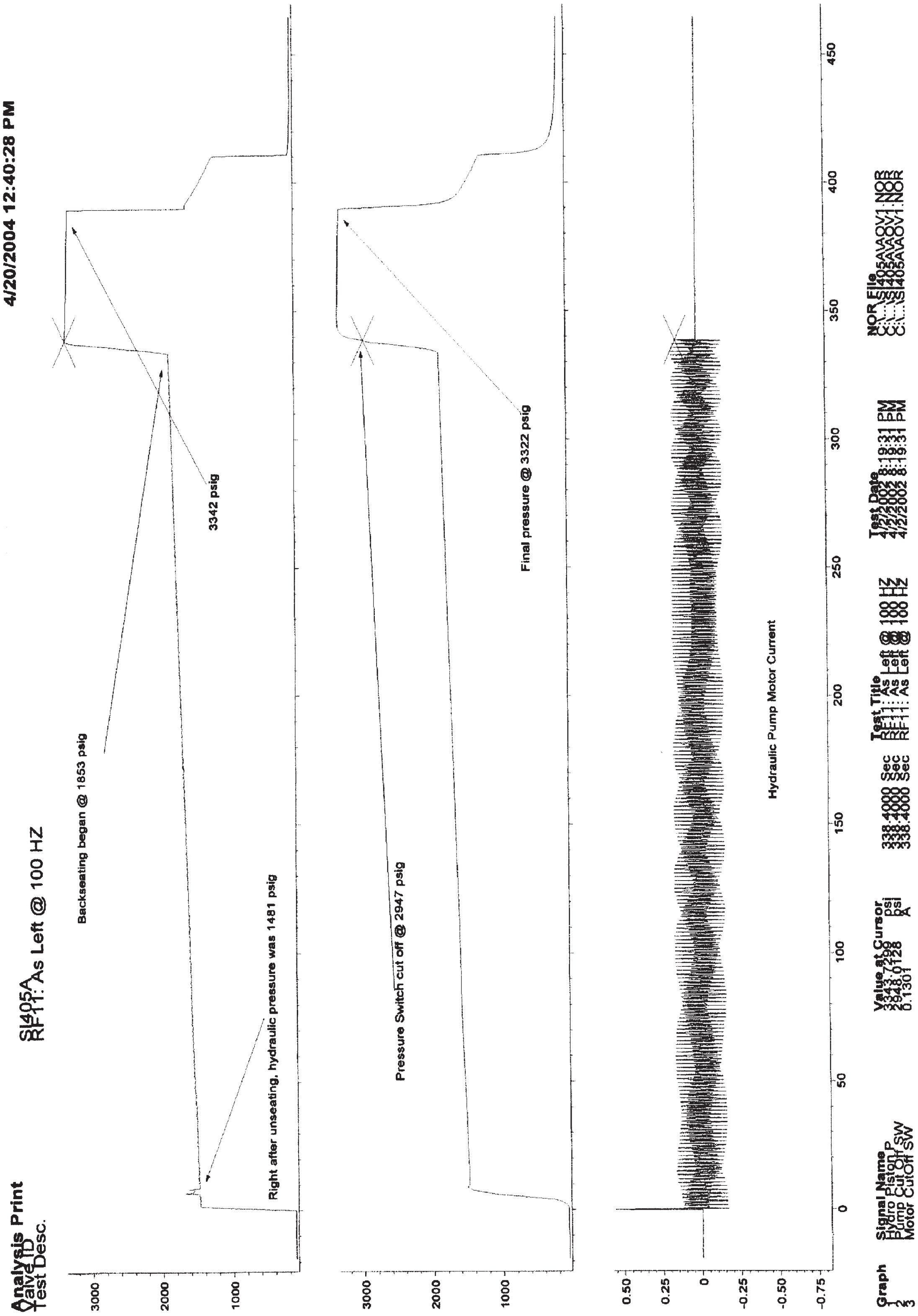


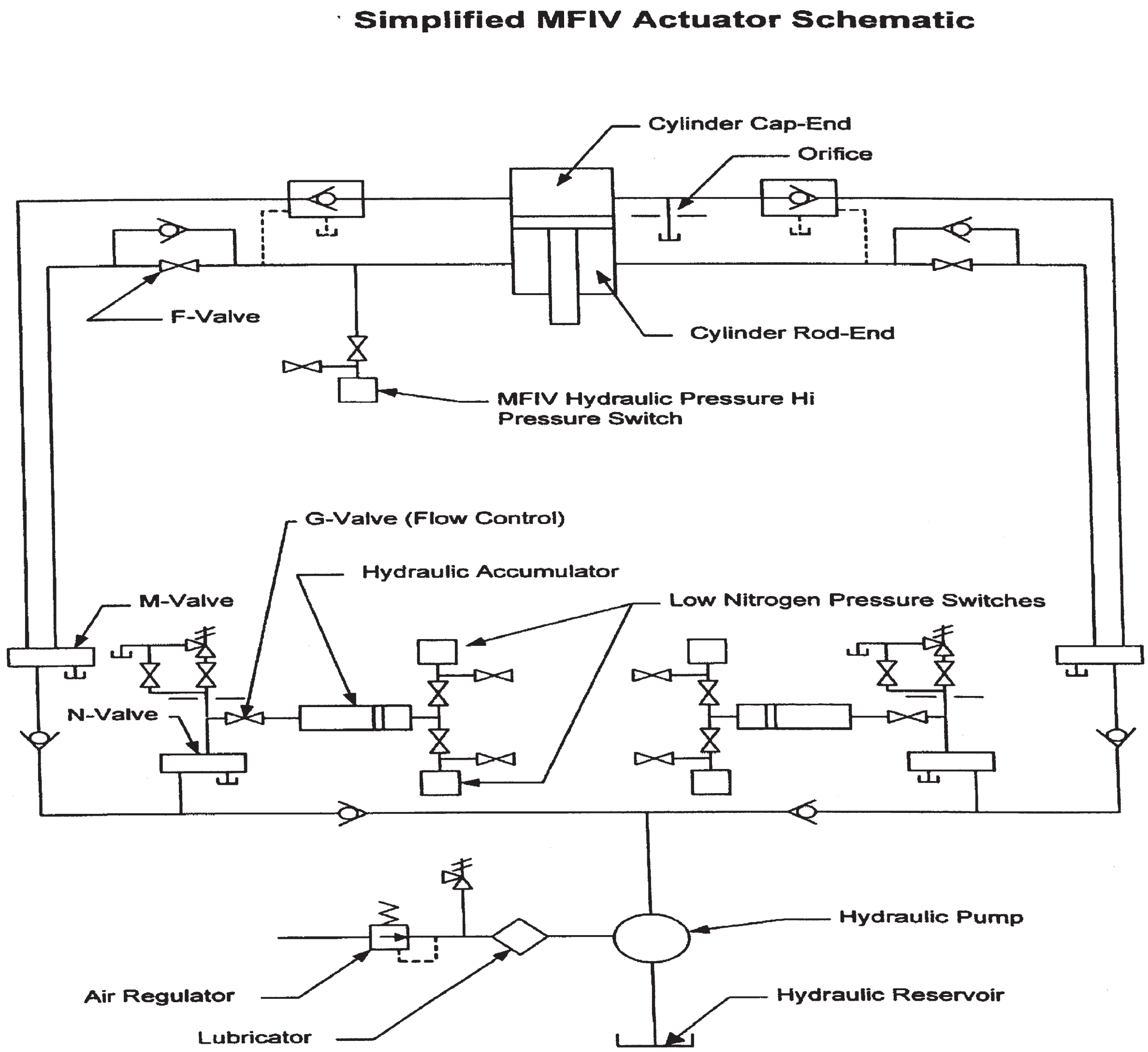
Shut Down Cooling "A" Valve
Before & After Nitrogen (Optimum) Settings

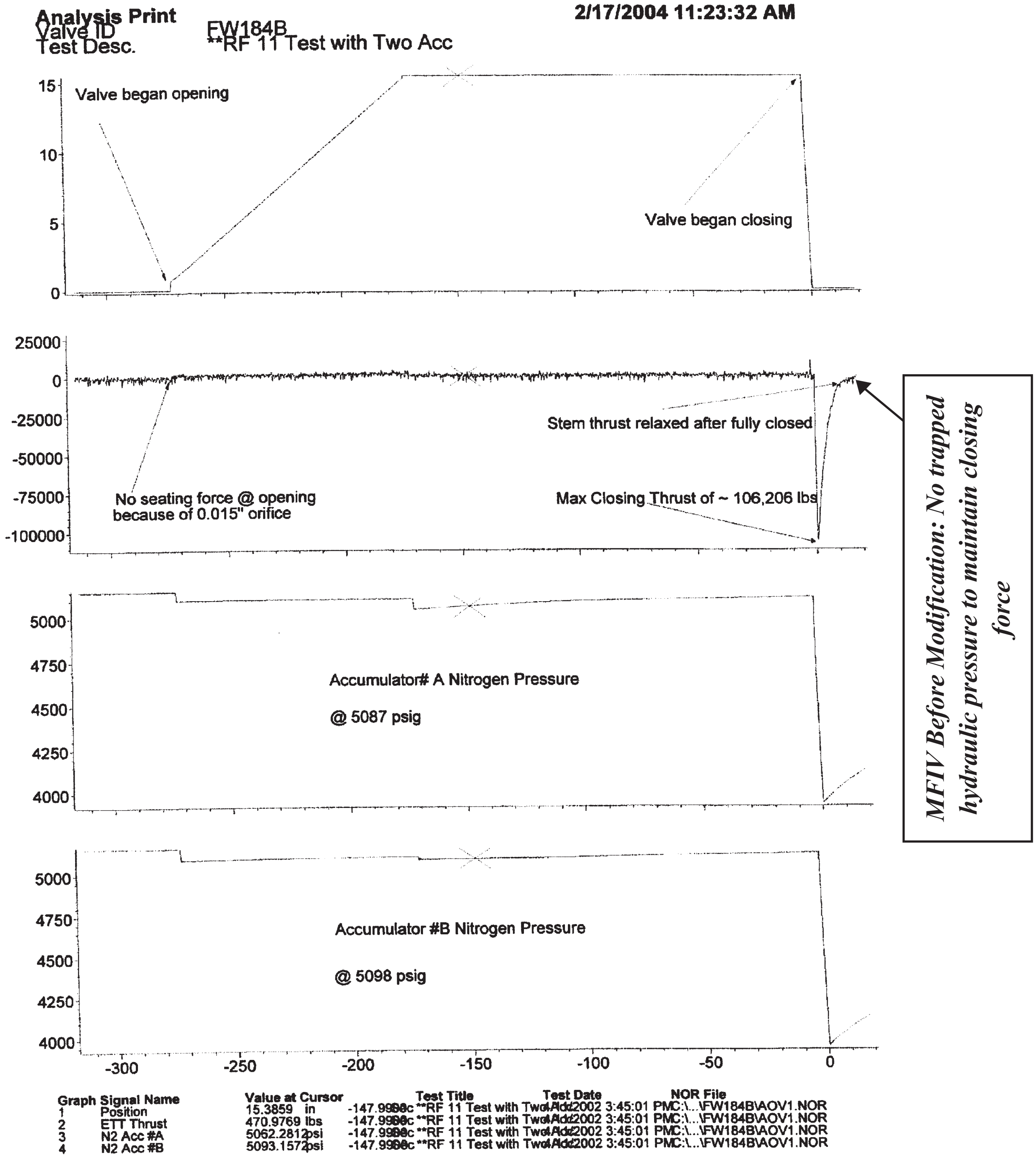


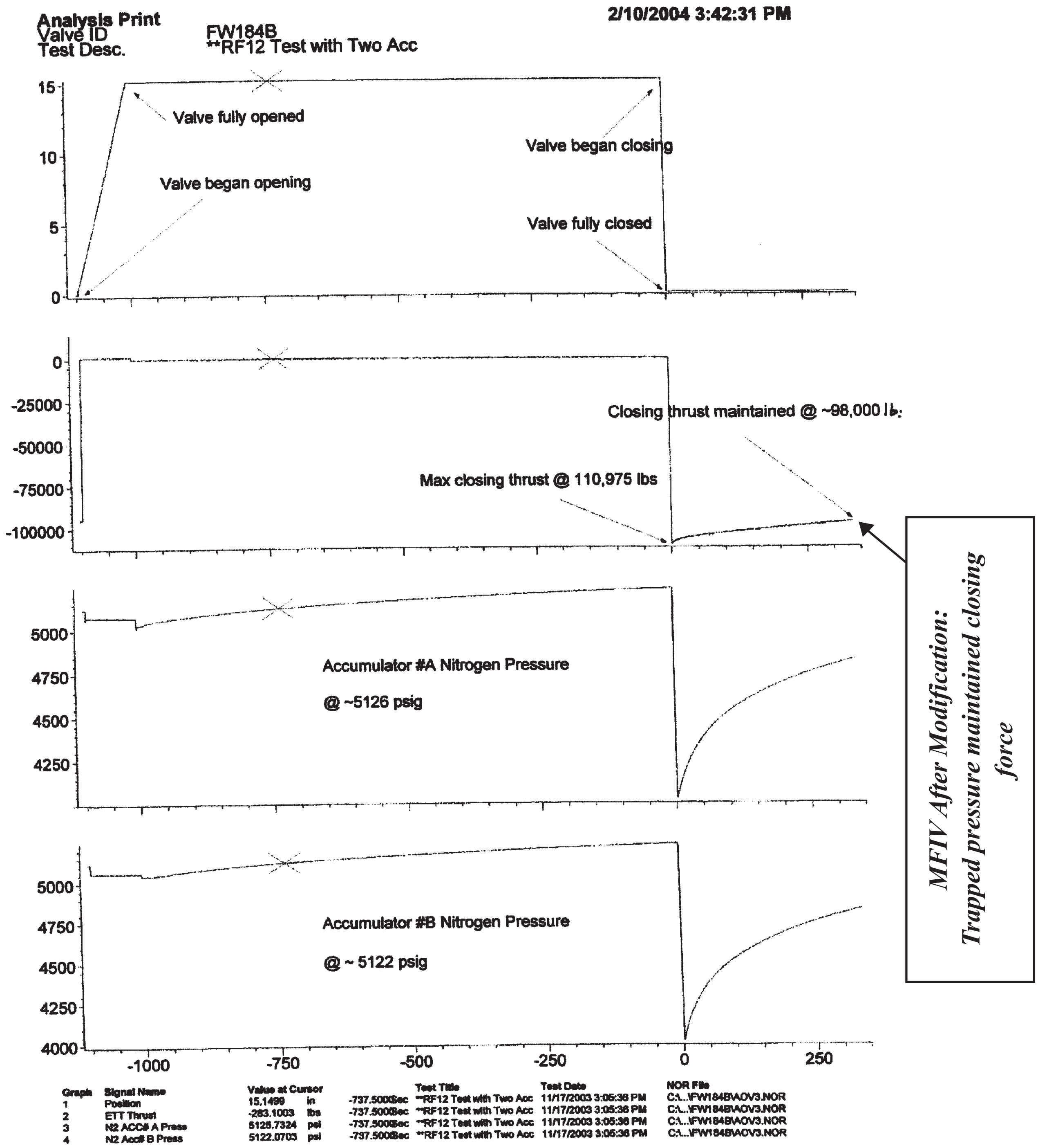
Analysis Print
Valve ID
SI405A
20000
0
-20000
0
50
100
150
200
250
300
350
400
At Nitrogen pressure of 1443 psig
1) Max Unseating Thrust of 14,456 lbs
2) Max backseating thrust of 18,304 lbs
At Nitrogen pressure of 1135 psig:
1) Max Unseating Thrust of 6550 lbs
2) Max backseating thrust of 34,625 lbs
Additional available open thrust of 16321 lbs (34625-18304)
After reducing nitrogen charged pressure from 1443 psig to 1135 psig.
Trace
1
2
Signal Name
ETT Thrust
ETT Thrust
Value at Cursor
-415.9154 lbs
-1040.0406 lbs
65.4700 Sec
65.4700 Sec
Test Title
RF11: As Left @ 100 HZ
RF11: AF Test3 @ 100 Hz
Test Date
4/2/2002 8:19:31 PM
3/30/2002 7:25:32 AM
NOR File
C:\...\SI405A\AAOV1.NOR
C:\...\SI405A\AAOV2.NOR


Analysis Print
Valve ID SI405A
Test Desc. RF11: As Left @ 100 HZ
Backseating began @ 1853 psig
Right after unseating, hydraulic pressure was 1481 psig
Nitrogen pressure @ 1433 psig
Right after unseating, nitrogen pressure was 1148 psig
Valve stem travel @ 11.114 inches
Valve stem travel @ 0.434 inch
Graph
1
2
3
Signal Name
Hydro Piston P
Nitrogen
Position
Value at Cursor
1853.5562 psi
1432.7472 psi
11.1139 in
331.8700 Sec
331.8700 Sec
331.8700 Sec
Test Title
RF11: As Left @ 100 HZ
RF11: As Left @ 100 HZ
RF11: As Left @ 100 HZ
Test Date
4/2/2002 8:19:31 PM
4/2/2002 8:19:31 PM
4/2/2002 8:19:31 PM
NOR File
C:\...\SI405AVAOV1.NOR
C:\...\SI405AVAOV1.NOR
C:\...\SI405AVAOV1.NOR
3000
2000
1000
1400
1300
1200
1100
10.0
7.5
5.0
2.5
0
0
50
100
150
200
250
300
350
400
450


Analysis Print
Valve ID SI405A
Test Desc. RF11: As Left @ 100 HZ
4/20/2004 12:40:28 PM
Right after unseating, hydraulic pressure was 1481 psig
Backseating began @ 1853 psig
3342 psig
Pressure Switch cut off @ 2947 psig
Final pressure @ 3322 psig
Hydraulic Pump Motor Current
Graph 1 2 3
Signal Name Hydro Piston P Pump Cut Off SW Motor CutOff SW
Value at Cursor 3343.7299 psi 3948.0128 psi 0.1301 A
338.4000 Sec 338.4000 Sec 338.4000 Sec
Test Title RF11: As Left @ 100 HZ RF11: As Left @ 100 HZ RF11: As Left @ 100 HZ
Test Date 4/2/2002 8:19:31 PM 4/2/2002 8:19:31 PM 4/2/2002 8:19:31 PM
NOR File C:\...\SI405AVAOV1.NOR C:\...\SI405AVAOV1.NOR C:\...\SI405AVAOV1.NOR

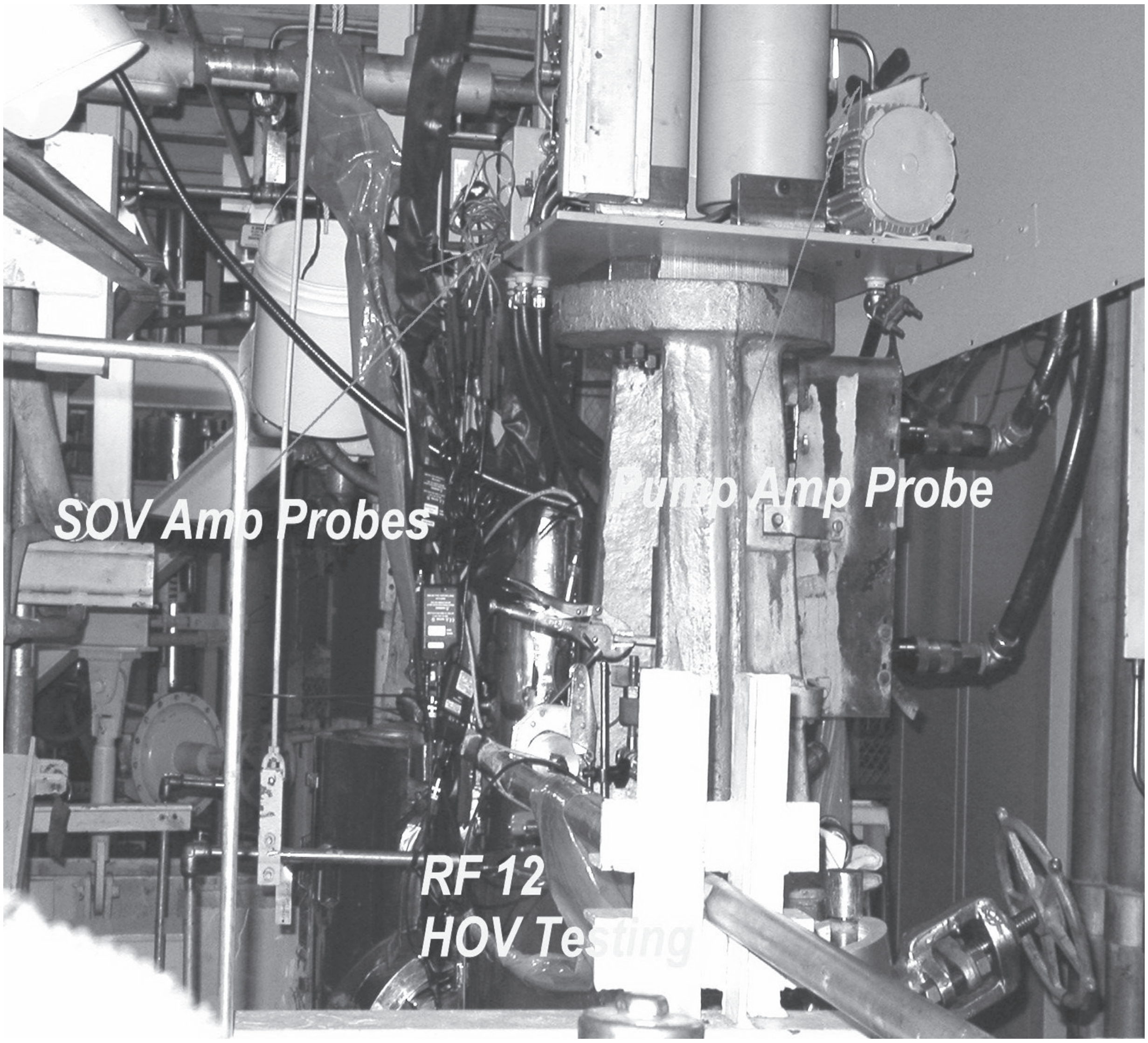

Simplified MFIV Actuator Schematic
Cylinder Cap-End
Orifice
F-Valve
Cylinder Rod-End
MFIV Hydraulic Pressure Hi
Pressure Switch
G-Valve (Flow Control)
Hydraulic Accumulator
M-Valve
Low Nitrogen Pressure Switches
N-Valve
Hydraulic Pump
Air Regulator
Hydraulic Reservoir
Lubricator


Analysis Print
2/17/2004 11:23:32 AM
Valve ID FW184B
Test Desc. **RF 11 Test with Two Acc
Valve began opening
Valve began closing
Stem thrust relaxed after fully closed
No seating force @ opening because of 0.015" orifice
Max Closing Thrust of ~ 106,206 lbs
Accumulator# A Nitrogen Pressure
@ 5087 psig
Accumulator #B Nitrogen Pressure
@ 5098 psig
MFIV Before Modification: No trapped hydraulic pressure to maintain closing force
Graph Signal Name
1 Position
2 ETT Thrust
3 N2 Acc #A
4 N2 Acc #B
Value at Cursor
15.3859 in
470.9769 lbs
5062.2812 psi
5093.1572 psi
Test Title
Test Date
NOR File

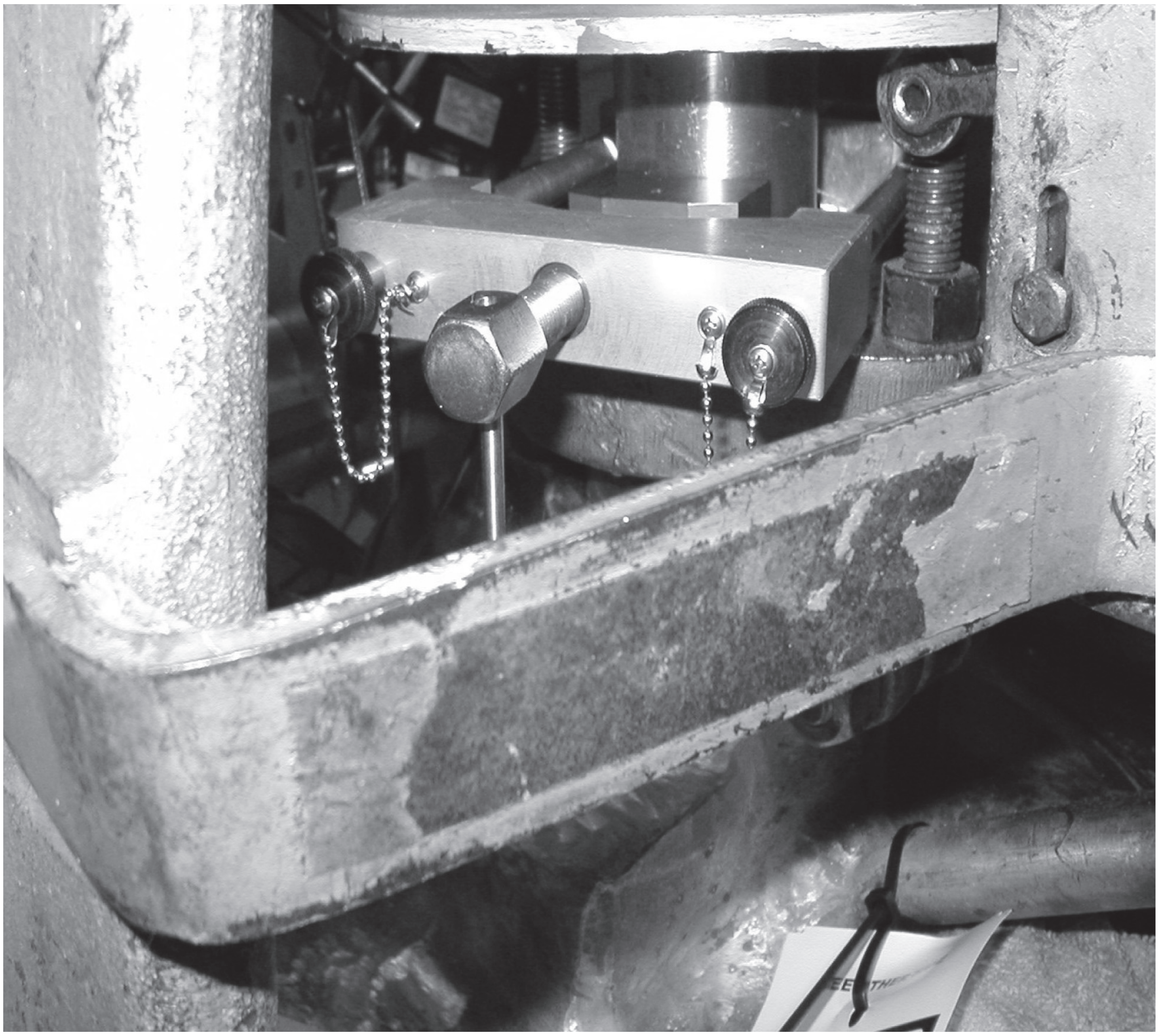

Analysis Print
2/10/2004 3:42:31 PM
Valve ID FW184B
Test Desc. **RF12 Test with Two Acc
Valve fully opened
Valve began closing
Valve began opening
Valve fully closed
Closing thrust maintained @ ~98,000 lb.
Max closing thrust @ 110,975 lbs
Accumulator #A Nitrogen Pressure
@ ~5126 psig
Accumulator #B Nitrogen Pressure
@ ~ 5122 psig
MFIV After Modification:
Trapped pressure maintained closing force
Graph Signal Name Value at Cursor Test Title Test Date NOR File
1 Position 15.1499 in -737.500Sec **RF12 Test with Two Acc 11/17/2003 3:05:36 PM C:\...\FW184B\AOV3.NOR
2 ETT Thrust -283.1003 lbs -737.500Sec **RF12 Test with Two Acc 11/17/2003 3:05:36 PM C:\...\FW184B\AOV3.NOR
3 N2 ACC# A Press 5125.7324 psi -737.500Sec **RF12 Test with Two Acc 11/17/2003 3:05:36 PM C:\...\FW184B\AOV3.NOR
4 N2 Acc# B Press 5122.0703 psi -737.500Sec **RF12 Test with Two Acc 11/17/2003 3:05:36 PM C:\...\FW184B\AOV3.NOR

Shut Down Cooling Isolation Valve

Test Sensors



Shut Down Cooling Isolation Valve
Removable ("D" Clamp) Strain Gauge



Main Feed Water Isolation Valve

Permanent Mounted Strain Gauge

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Exelon Nuclear MOV Program Standardization

17 Units, 10 Stations and 1 Best MOV Program

Ted Neckowicz
Steve Gallogly
Exelon Nuclear

The Objective

In November 2002, Exelon Nuclear rolled out its standardized Motor-Operated Valve (MOV) Program to all 10 sites within the Exelon/Amergen fleet. This MOV Program Standardization, which we believe to this day, is the most comprehensive valve program change anywhere in the nuclear industry. The MOV Program changes involved 17 separate MOV procedures and Guidelines (we call them T&RMs) and common centralized software that integrate the procedures and guidelines into one standardized process. Given that the changes involved were complex and had potential significant station impact, a formal project was established with periodic progress and management report outs. A three-man core team provided the foundation of the project with one serving as the Project Manager. The project work was done as level of effort with the project core team fulfilling their normal responsibilities. While the project had several significant challenges and was delayed four months from the schedule originally planned, management sponsorship and focus on the ultimate goal lead to the project success. Now Exelon Nuclear's MOV program is well positioned to reap the benefits of the standardization effort which include effective resource sharing, remote off-site support, reduction of human errors, "state of the art" set-point management /configuration control and improved MOV reliability at a reduced implementation costs. Future program maintenance is also reduced given that only one MOV program rather than 10 site-specific programs exist. Borrowing the famous line, Exelon's MOV Program can now proudly say it's "All for One – One for All".

Who is Exelon Nuclear

Exelon Nuclear is made up of the 5 former ComEd Nuclear Stations including Byron, Braidwood, Dresden, LaSalle and Quad Cities, 2 former PECO Energy Stations including Limerick and Peach Bottom, 2 former GPU stations including Oyster Creek and Three Mile Island, and finally Clinton Station formerly of Illinois Power. These companies were combined to form Exelon in 1999.

The Call to Standardize

At the end of 2000, the call to standardize the Exelon MOV Program was actually part of a much bigger initiative to standardized company wide processes and programs inside and outside of Exelon Nuclear. A Chief Executive Officer (CEO) level corporate commitment to Wall Street proclaimed that Exelon would standardize all business units by the end of 2002. This commitment was the source of the High Level executive sponsorship that became invaluable as various obstacles were encountered. Each engineering program was selected and prioritized by upper management for standardization, with the MOV Program rated as one of the most difficult engineering program given the high level of institutionalization and regulatory oversight. The MOV program was given an original standardization deadline of 6/30/02; one of the last engineering programs. This later changed to 10/31/02 due to project delays. Nonetheless, the project successfully fulfilled the corporate standardization commitment.

The first meeting to conceptually design Exelon's MOV Program Standardization was held during the January 2001 Motor Operated Valve Users Group Meeting in Clearwater. Key participates at that meeting included Ted Neckowicz (former PECO & current Mid Atlantic MOV Engineer), Steve Gallogly (former PECO & current Mid Atlantic Valve Maintenance Specialist), Brian Bunte (former ComEd MOV Engineer) and Bill Cote (current Mid-West MOV Engineer). Each person independently ranked what program attributes they believed would be most beneficial to standardize under the new standardization initiative. Needless to say, this process identified considerable differences in viewpoints between the group members that they were challenged to resolve in order to formulate the initial Standard MOV Program Development Strategy. While initially highly dynamic, this strategy ultimately can be summarized as follows:

- Adopt a best practice approach based on technical merit not on "this is how we do things here at [pick a site…]"

- Design a process that accomplishes the shift from NRC Generic Letter (GL) 89-10 "justify engineering assumptions" to GL 96-05 performance monitoring
- Provide maintenance personnel with simplified criteria that makes MOV diagnostic testing as much like performing a routine surveillance test as possible
- Fully integrate a testing, trending and design into a common process
- Provide procedural guidance to minimize the need for "tribal knowledge" and to achieve consistent test guidance
- Focus on processes and common implementation tools instead of testing hardware and implementation minutiae
- Design fully integrated engineering and maintenance software that is accessible from any computer with access to the Exelon intranet
- Create a simple software interface that is user friendly to less computer savvy maintenance personnel
- Implement common quantitative MOV program performance and health indicators

Quickly this informal program strategy lead to the next step, the development of the formal project plan.

The Project Plan

The Project Plan was written over a period of several days by Ted Neckowicz and Bill Cote who were the principal leads for the engineering initiative, thus the project nick name became "Bill and Ted's Exelon Adventure". The Project Plan discussed the following:

1. Program/Process Ownership
2. Project Strategy
3. Interfaces and Control
4. Implementing Procedure Hierarchy
5. Project Phases
6. Budget
7. Baseline Schedule
8. Exceptions to Standardization
9. Site Program Transition
10. Critical Success Factors
11. Management Reporting

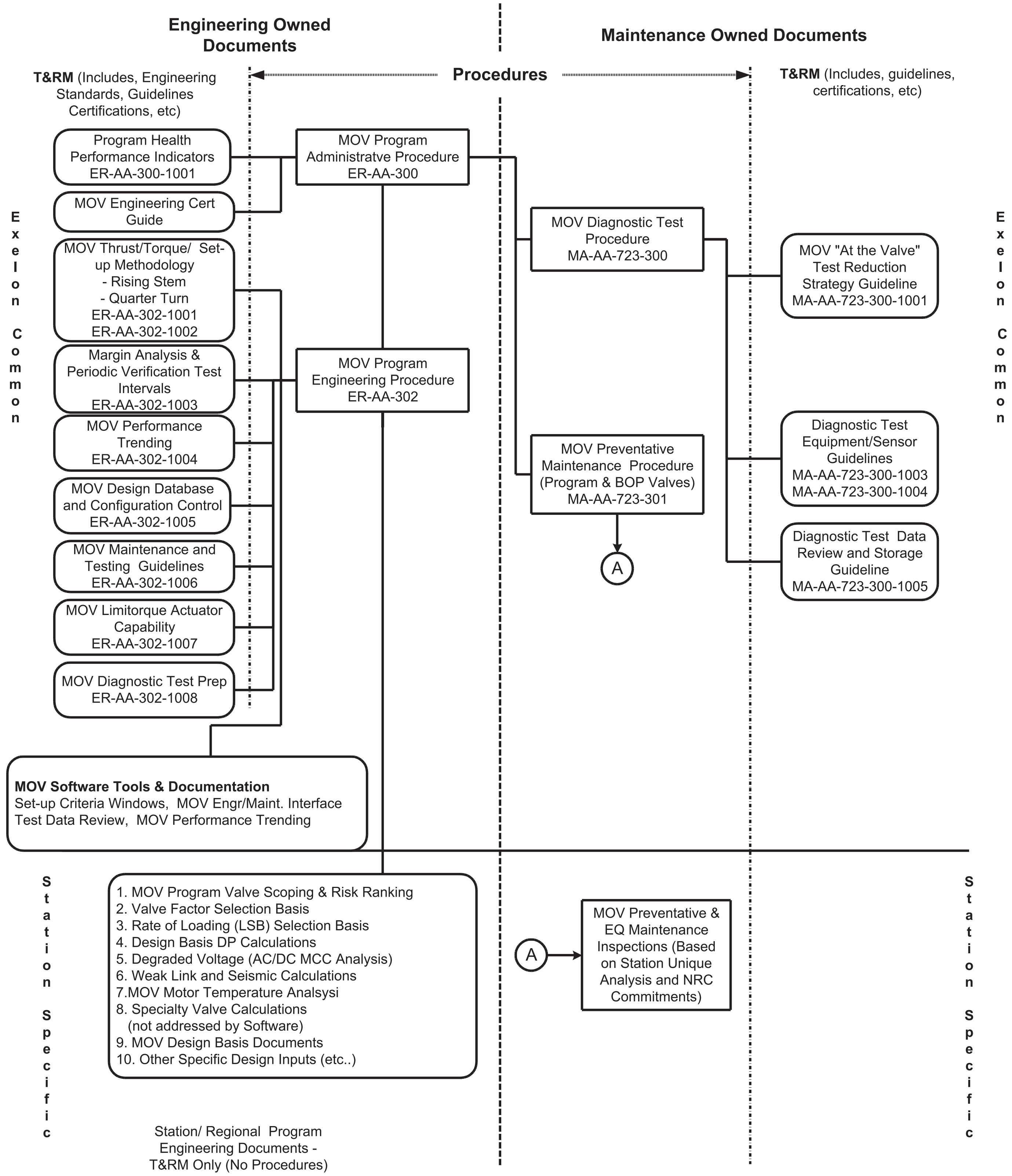
The project plan strategy proposed the following key standardized elements:

- A standardized methodology and calculational software to execute MOV Calculations and manage engineering data.
- A three (3) step MOV Test management process to be facilitated by new software to be developed that includes: Test Preparation, Data Review and Trending.
- A standardized MOV Data Analysis platform to review and store MOV Diagnostic traces. Quiklook for Windows was selected based on ability to process both VOTES and Quiklook data.
- A "Maintenance-owned" testing process where qualified MOV Maintenance Technicians can conduct all routine in-plant MOV diagnostic testing and test acceptance for returning the MOV back to service (operable) without "at the valve" MOV engineer involvement.

Through implementing these standardized elements, the core group believed that Exelon would reap the best long-term MOV Program efficiency gains.

The project plan identified the following (17) new Engineering and Maintenance Procedures and T&RMs for development (See Figure 1).

Figure 1
Exelon MOV Program
MOV Program Procedure / T&RM Heirarchy



Engineering Procedures

1. Motor Operated Valve Program Administrative Procedure
2. Motor Operated Valve Program Engineering Procedure

Engineering Technical & Reference Material (T&RM)

1. Rising Stem Motor Operated Valve Thrust & Torque Sizing and Set-up Window Determination Methodology
2. Quarter-Turn Motor Operated Valve Sizing and Set-up Window Determination Methodology
3. MOV Margin Analysis and Periodic Verification Test Intervals
4. Motor Operated Valve Performance Trending
5. Motor Operated Valve Design Database Control and Design Datasheet Activities
6. Motor Operated Valve Maintenance and Testing Guidelines
7. MOV Limitorque Actuator Capability Determination Methodology
8. MOV Diagnostic Test Preparation Instructions
9. MOV Program Performance Indicators

Maintenance Procedures

1. MOV Diagnostic Test Procedure
2. MOV Preventative Maintenance Procedure

Maintenance Technical & Reference Material (T&RM)

1. MOV "At The Valve" Diagnostic Test Reduction Strategy
2. VOTES Diagnostic Test Equipment / Sensor Guideline
3. QUIKLOOK Diagnostic Test Equipment / Sensor Guideline
4. Review and Evaluation of Motor Operated Valve Test Data

MOV Program attributes that were excluded from MOV Standardization included:

- MOV Diagnostic Test Data Acquisition Equipment – Diagnostic Test data acquisition equipment was not standardized due to the high implementation cost for 10 sites. The Test Analysis Platform was standardized regardless of the diagnostic test acquisition system (i.e. VOTES, QUIKLOOK).
- Valve Factor and Rate of Loading basis – These values are all considered embedded to the site-specific GL 89-10 closure requirements. Very limited program efficiency gain.
- Design Basis Bounding Stem Factor basis – These values are considered embedded to site specific GL 89-10 closure requirements and stem lube type and maintenance practices. Very limited program efficiency gain.
- No Program scope changes were made nor were any MOV design basis reviews revisited as part of MOV Standardization.
- MOV Risk Ranking methodology was standardized using NRC approved methodology. Risk rankings were not immediately revised; however, MOV risk rankings are to be reviewed and adjusted during required periodic site Probabilistic Risk Assessment (PRA) updates.

Project Phases

Project Development – Develop Project Plan (See above).

Procedure Development – The project core team was comprised of Project Manager, Ted Neckowicz (Mid Atlantic – MOV Program Engineer), Bill Cote (Mid-West – MOV Program Engineer) and Steve Gallogly (Corporate Valve Maintenance Specialist). Each Core team member had responsibility for the development of a specific number of draft documents as level of effort activities. Another core team member then reviewed each draft. Following this, each draft went through the following rigorous document review process:

- Site Subject Matter Expert (SME) Review Cycle
- Site Functional Area Manager (SFAM) Review Cycle
- Fatal Flaw Review Cycle
- Corporate Functional Area Manager (CFAM) Review
- Site Approval & Implementation

Each procedure was tracked on a resource-managed schedule. Resources were shifted and all other work except critical support of plant emergent issues was delayed, as necessary, to keep the procedure schedule on track. The MOV Program documents were ready for site approval by the end of June 2002. The procedures were to be implemented in conjunction with the deployment of the MIDAS software later in the fall.

Software Development – New Quality Assured Software was to be developed to implement the new MOV Program process including the standardized sizing methodology. Because of the best practice approach to the software development, all stations had some changes to their existing MOV set-point calculations requiring validation. Additionally, the 3 Step MOV Test Management software process was new to every Exelon station.
Software development started in early 2002 when the 2002 engineering project budget became available. Based on review of existing MOV software products available both internal and external to Exelon, a decision was made to modify the existing PECO MOV software, which was deployed at the PECO plants in 2000. Teledyne Instruments had developed the "MIDAS for Windows" for PECO converting PECO's DOS based MIDAS MOV sizing software to a Windows 2000 GUI based software product. At the time, general consensus of the Exelon MOV subject matter experts was that "MIDAS for Windows" was the most technically advanced and best product available to further modify to support Exelon Standardization.

The MOV program documents provided most of the technical basis for what the new standardized software did and how it did them. Project schedule requirements required several months of overlap between MOV document completion and software development. This posed a significant challenge to Teledyne who was initially developing software based on documents that were frequently changing. This issue was managed only through close coordination and frequent communication between the Exelon Project Manager and Teledyne Instruments. Teledyne Instruments, in particular Michael Richard, played a critical role in making the software development a success through their high level corporate commitment to the project.

Two MOV software products were developed: MIDAS and MIDATEST

MIDAS – MIDAS is the primary MOV engineering tool that provides MOV design/sizing analysis, thrust/torque set-point methodology, margin analysis, PVT-interval analysis and configuration control. MIDAS MOV data are stored in a one record per MOV.

MIDATEST – MIDATEST is the primary MOV engineering and maintenance tool that provides 1) MOV Diagnostic Test Preparation, 2) Diagnostic Test and PM Data review and 3) MOV Data Analysis and Trending. MIDATEST MOV data are stored in a one record per Test/Work Order.

The MIDAS program was essentially complete by the mid-September 2002. Software V&V by Teledyne took nearly one month followed by Exelon acceptance testing. With the availability of an approved MIDAS, the standard MOV Program rolled out on schedule to the 10 sites and 2 corporate offices on October 31st 2002. This included conversion of all existing MOV data into the new MIDAS format and providing Citrix access to the primary software users in both Engineering and Maintenance at all sites.

Program Implementation and Transition Period
Site-specific implementation dates were established at or after the corporate process rollout on 10/31/02. Stations without near term refueling outages began implementing the process the week of 11/03/02.

Implementation Date: The site specific date after which all new MOV Program activities will be started using the new Exelon standard MOV Program. Activities include MOV set-up window calculations, margin review, MOV test package preparation, diagnostic test review and MOV performance trending.

Transition Period: The period following the implementation date during which MOV testing activities initiated under the former program will be completed (e.g., tested and reviewed) using the same (i.e. former) program. This transition period will be nominally twelve weeks based on the T-12 work planning process.

> ***MOV Program Transition Period Example***
>
> *Scenario* - Limerick implements the new program on 10/31/02 and has an April 7, 2003, outage with on-line MOV work scheduled in November, December 2002 and January 2003.
>
> *Acceptable Limerick Transition Plan* - MOV testing scheduled for 11/02 through 1/03 and previously planned using the existing program before 10/31/02 may be completed using the existing program. All new MOV calculations and test package preparations required for the April 2003 refuel outage and for on-line testing 12 weeks after 10/31/02 shall be prepared using the new MOV Program process. Any new MOV calculations and test package preparations prepared after 10/31/02 shall be done using the new MOV Program process.

Change Management
With a project of this size and affecting 10 stations and 2 corporate offices, a change management plan was required. The change management plan was periodically reviewed by management and rolled out to each of the sites. The change management contained the following:

> Site Implementation dates (based on Fall/Winter Outage conflicts)
>
> Barriers to success – Plans to address

Corporate Actions required to Implement Program (See Example in Table 1)

Site Actions required to Implement – A 2 year implementation period was specified to convert and approve all existing program MOV calculations to the new MIDAS software.

Table 1 (Typical Corporate Implementation Actions)

Task Description	Target Date
Develop and verify MIDAS Software	8/30/02
Complete IT MIDAS software requirements	9/13/02
Develop and verify MIDATEST Software	9/30/02
Complete IT MIDATEST software requirements	10/13/02
Process and Software Training Development	8/15/02
Provide Process Training to MWROG Engineering	9/01/02
Provide Process Training to MWROG Maintenance	9/01/02
Provide Process Training to MAROG Engineering	9/15/02
Provide Process Training to MAROG Maintenance	9/15/02
Quiklook Diagnostic Analysis Training	9/30/02
Quiklook Software IT requirements complete	9/30/02
Assist with Site Data Migration and IT Start-up Support	10/1-31
Supersede or revise corporate level engineering documents	11/30/02
Implement Revised MOV Program Engineering Cert Guide	10/31/02

Training

As indicated above in Table 1, several training sessions were arranged in both the Mid-Atlantic and Mid-West Regions prior to the implementation date. Formal Lesson Plans were developed including practical factor exercises and exams. The training focused primarily on using the new software, which was new to all 10 Exelon sites. Follow-up training is routinely provided after the implementation date using Web training tools such as NetMeeting.

The Keys to Success

Looking back at the project and the barriers encountered, several essential keys to the project's success are noteworthy. They include:

- Senior Management was absolutely committed to successful Standardization implementation. If a specific station or corporate workgroup refused or not adequately support the project, their organization would soon hear from the senior management.
- New procedures and processes were developed by a small core of individuals and presented to the 10 stations for review and comment. "Management by committee" was minimized.
- Once the comment period expired and the comments were dispositioned, only a "Fatal Flaw" identified by a station could prevent approval and implementation. This eliminated the continual cycling of a procedure to incorporate late comments.
- The Citrix server based deployment allowing centralized (single) software installation. This deployment strategy eliminated the need for software installations on every user's personal computer and eliminated the compatibility and software QA problems inherently created. MIDAS has over 120 users throughout Exelon and that list still continues to grow. Without this deployment strategy, the project could not have succeeded.

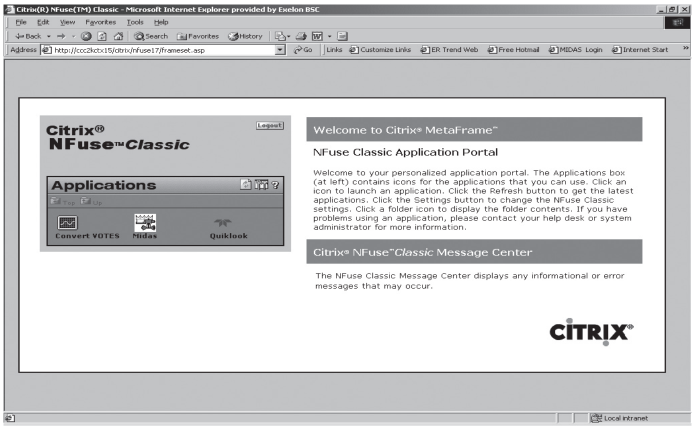
Continual Improvement – Effectiveness reviews

Even with the best of intentions and planning, it was anticipated that some changes or additional enhancements would be necessary to effectively implement the new MOV Program. Consequently, the project had planned and budgeted in 2003 for a program effectiveness review and for additional software improvements. The effectiveness review was conducted during the 2nd quarter of 2003 and the software upgrades rolled out in November 2003.

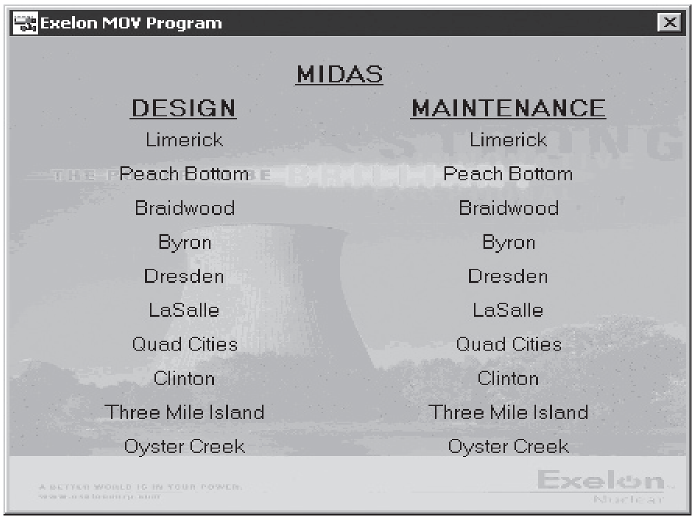
MIDAS & MIDATEST

The Software that makes it all work

The three standardized MOV Program software applications are all accessible via Microsoft Explorer via a Citrix application server and can be accessed from any computer connected to the Exelon intranet.

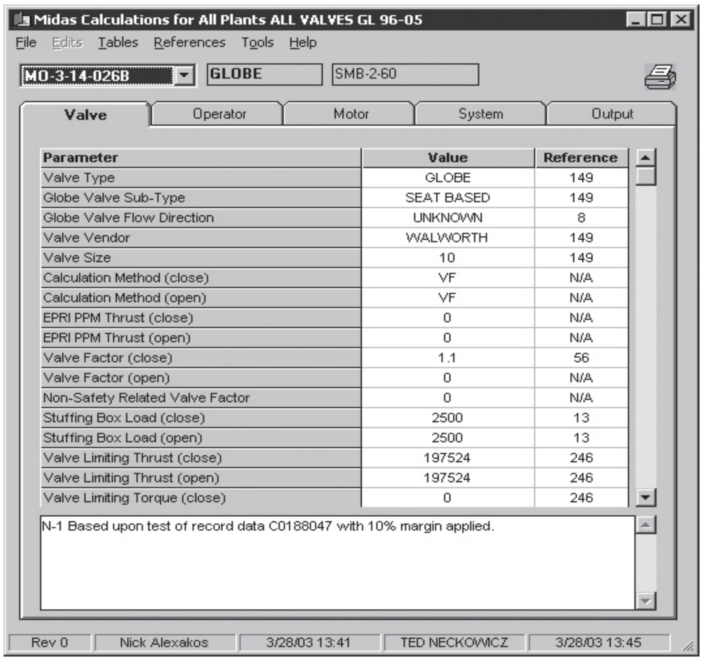


Selecting MIDAS on the Citrix screen runs the MIDAS/MIDATEST launch pad program. Either MIDAS (Design) and/or MIDATEST (Maintenance) launches when the appropriate site database is selected. Any authorized user can access and view any site database. Different levels of edit privileges can be set for each user.

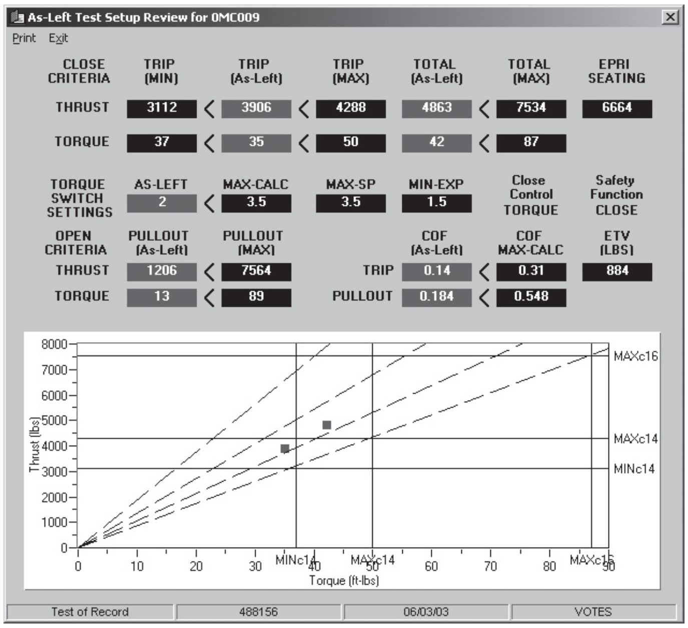


MIDAS

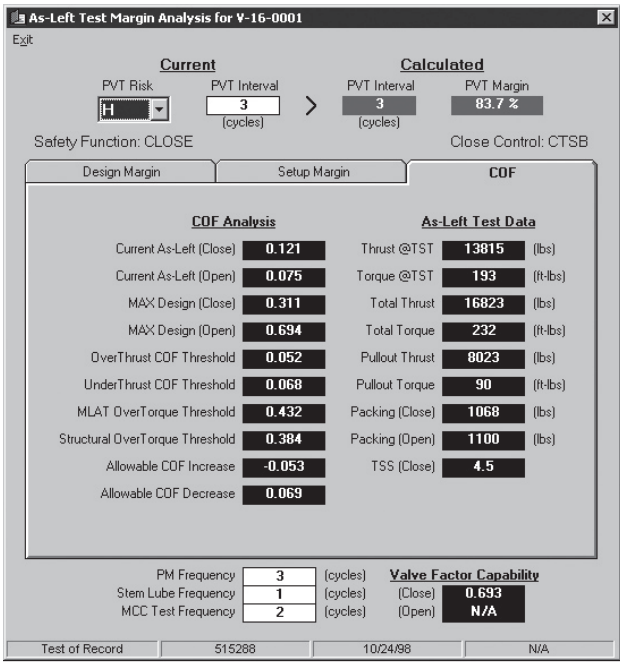
The basic MIDAS interface and main form is shown below. The screen shows an approved Peach Bottom MOV Design Data Record. The revision level, preparer, checker and approval date are shown on the status bar at the bottom.



The screen below shows the resulting set-up window criteria and the current Test of Record Data for a Clinton MOV. MIDAS stores the current Test of Record data in order to perform margin reviews.



The margin tables are displayed below for an Oyster Creek Valve. MIDAS performs set-up margins, design margins and stem COF analysis to assess each valve. Depending of safety function direction, control scheme and valve type, the appropriate margins are combined to determine the PVT margin used to establish the maximum test interval. Additionally, valve factor capability is calculated.



Other MIDAS capabilities include:

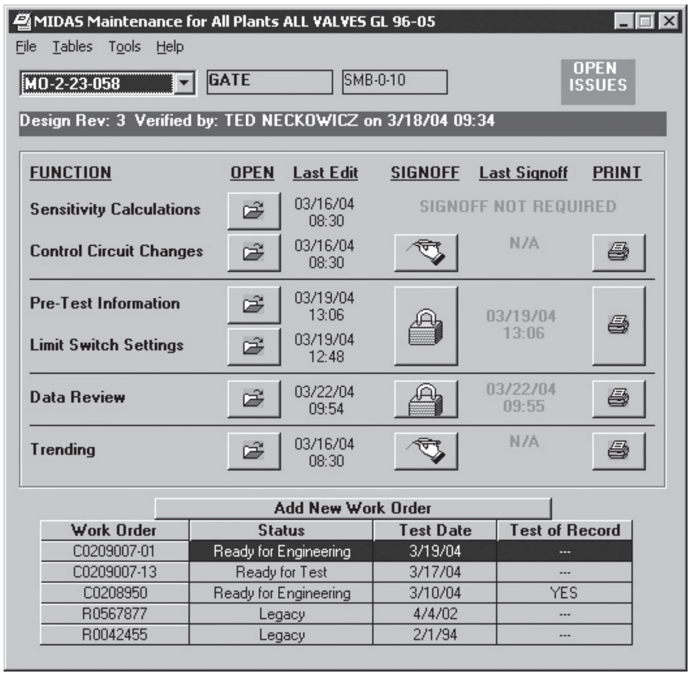
- MOV Voltage drop analysis
- ComEd AC Motor Methodology
- BWROG DC Motor Degraded Stroke Time Analysis
- EPRI Butterfly Torque Methodology
- EPRI Unwedging Analysis
- Powerful Export to Excel Reporting Tool
- Global Parameter Evaluator

MIDATEST

Shown below is the main MIDATEST screen. It shows the available test records in the grid at the bottom of the screen. A new record is created for each new diagnostic testing work order.

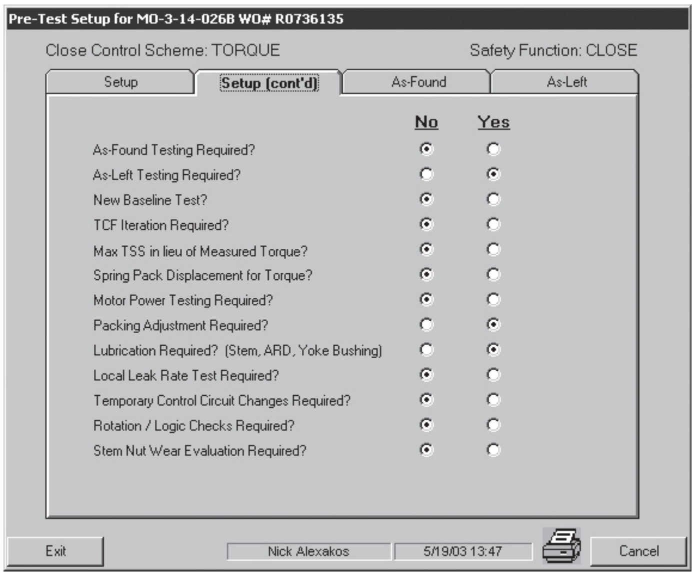
The current MIDAS record status shows up in the status bar. Only approved MIDAS design records are available for use in MIDATEST.

Each module of the MIDATEST software has individual signoffs. Status changes as the valve moves through the testing process from Pre-test to Data Review and then to Trending as each stage is signed off. The current record is shown as complete. Consequently, the Pre-Test, Data Review and Trending are all signed off and locked.



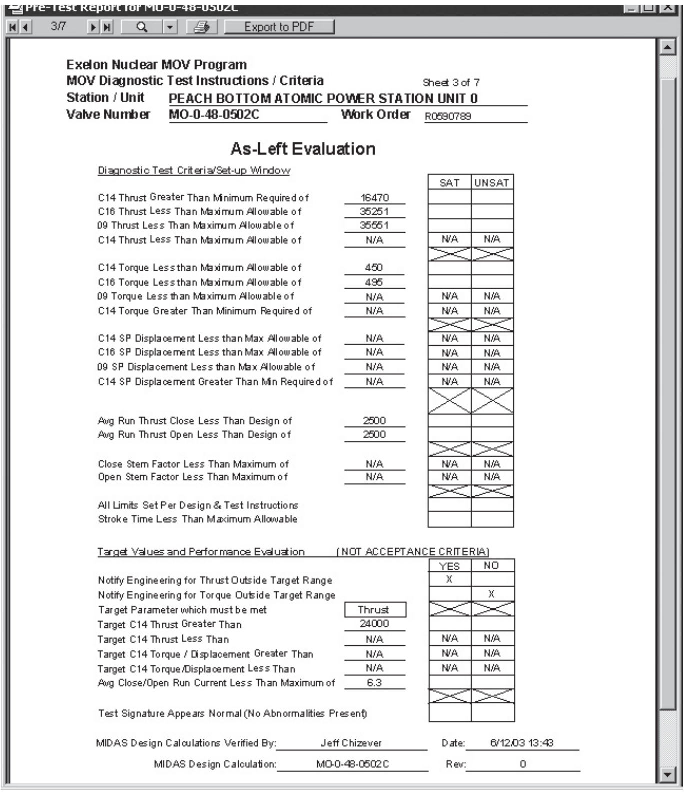
Pre-Test Instructions

Menu Driven Software Guides the Engineer Through the Pre-Test Preparation Process. Each software step in the decision making process is provided with procedure guidance and examples.



Maintenance Instructions are formatted to facilitate a Pre-Job Brief.

- A simple format is used on the first page of the test instructions to communicate general test requirements.
- **Only required test acceptance criteria are provided to maintenance (e.g., Standard (i.e. Thrust and Torque) or Thrust Only or Torque Only).**
- The Diagnostic Test Criteria/Instructions are structured to minimize the potential errors and confusion during testing (e.g., the software will "N/A" information that is not required in advance of the procedure going to the field). (See sample printout on next page.)

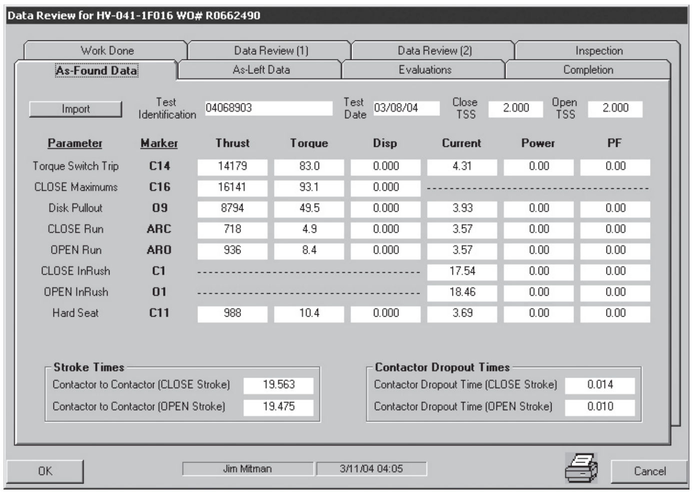


MOV Diagnostic testing is performed with a common procedure utilizing the Pre-Test Instructions

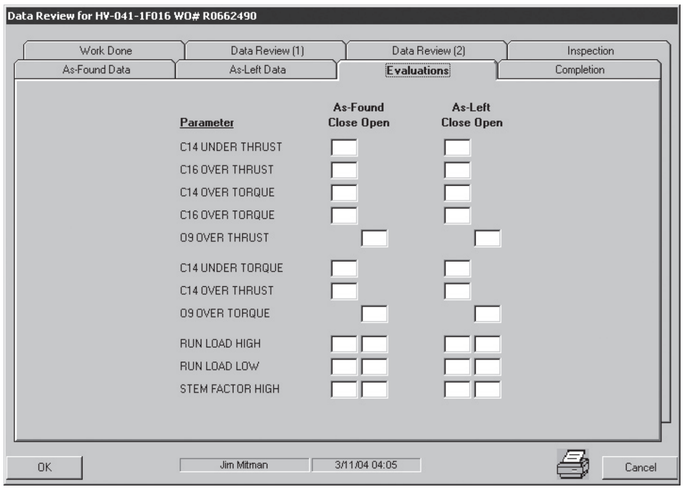
- The test procedure is designed to minimize or eliminate the redundant recording of data.
- The test instructions are included as part of the permanent test record.
- Numerical test results are not required to be transcribed into the procedure.
- As Left test results are independently verified.
- If all Test Acceptance Criteria is satisfactory then the test is acceptable and the valve can be returned to operations at this time without additional review by engineering.

Test Data Documentation / Review - Menu Driven Software Guides Maintenance Through the Data Review Process

- Each software step is provided with procedure guidance and example.
- As-Found and As-Left test data results can be directly imported into the software to eliminate data entry errors. See as-found data entry screen below.



- Results are automatically compared with test criteria and flagged for disposition / errors. Obviously, no flags (shown with an X) are the preferred result.

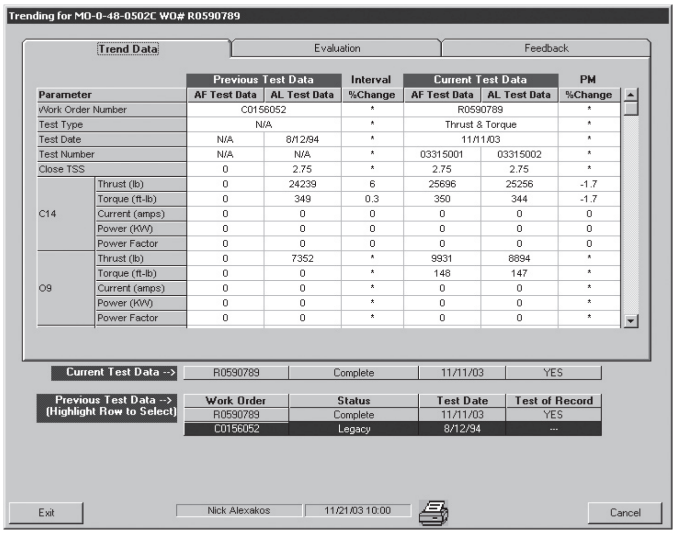


Maintenance Completes the Test Data Review

- Designation of "Test of Record" flags MIDAS that new "Test of Record" data is available for update in MIDAS.
- Once Engineering updates MIDAS with the new "Test of Record" data, all MOV margin evaluations will be based refreshed.

MOV Performance Trending

- Engineering Performs the Trending Review
- As Found test results for the current test are compared to the previous as left test results



- The change form as found to as left performance is also compared
- Quality of the test data for trending is confirmed
- Test performance is evaluated
- Engineering is required to evaluate if adjustments to the PM interval, Test interval or degradation factors in the design calculation prior to closing the trending module
- Engineering Completes the Trending Module and the Testing Process is Complete. Signoff of the Trending Module locks down the file and completes the testing process for the valve under the existing work order

MOV Program Health Reporting and Performance Indicators

Quarterly MOV Program Health Reports are prepared for each station in accordance with Exelon's procedure for management of Engineering Programs. In addition, quantitative Performance Indicators (PIs) are used to monitor the health of the MOV Program. Several of these performance indicators provide evidence of the material condition health and set-up margin. Additional performance indicators monitor the effectiveness of MOV periodic verification, preventative maintenance work activities, and associated recurring task frequencies. Lastly, other performance indicators monitor compliance with applicable GL 96-05 schedule commitments.

Performance Indicator Criteria are developed for the following Program attributes.

MOV Functional Failures (includes maintenance preventable, direct and indirect)

MOV Set-up Non-Conformance Conditions

MOV Margin

MOV Work Planning

MOV Diagnostic Test Proficiency

MOV Data Review

MOV Program Commitments

Emergent Industry/Regulatory Issues

Using the same technique used by the Exelon System Status Health Rating Guide, the following four MOV Program ratings will be established:

Each station is responsible for documenting the station specific MOV PI(s) that will be reported in the quarterly MOV program health reports.

MOV Program Performance Indicator Rating Criteria

White Rating Criteria (Sample)

Acceptable Functional Failure PI.

AND Acceptable Continuing and Singular Program Commitment PIs.

AND No more than two of the following PI(s) with Unacceptable Performance:

MOV NCC MOV Planning Test Proficiency

MOV Margin MOV Data Review

AND White or Green Emergent Industry/Regulatory Issue PI.

A Sample Station MOV Performance Indicators follows:

Rating Color	Performance	Action
Green	Excellent	Requires No Additional Attention at This Time
White	Acceptable	Current Performance and/or Activities are Acceptable
Yellow	Needs Improvement	Needs Additional Attention
Red	Not Acceptable	Risks High and/or Requires Excessive Monitoring/Resources to Maintain

ATOMVILLE

MOV Program Performance Indicators

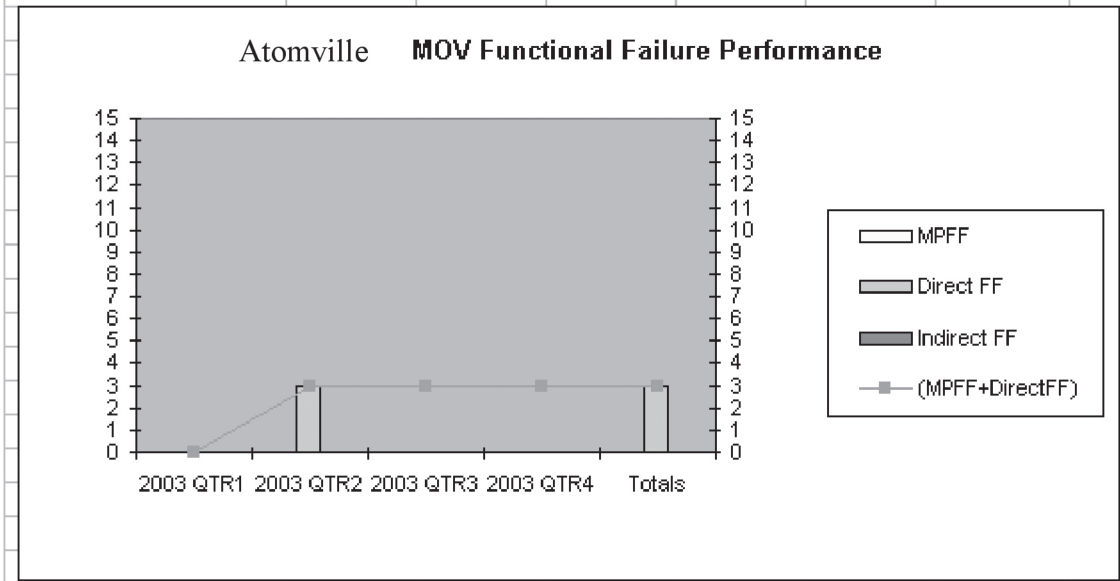
Overall MOV Program Performance - Needs Improvement

MOV Functional Failures	Unacceptable
MOV Non Conforming Conditions	Acceptable
MOV Margin	Unacceptable
MOV Work Planning	Acceptable
Diagnostic Test Proficiency	Acceptable
MOV Data Review	Acceptable
Commitments	Acceptable
Emergent Issues	Unacceptable

MOV Functional Failures - Unacceptable

Criteria: <= 1 MPFF per year/unit, <= 2.42 Direct FF per year (within scope of program control)

Trend Indirect FF (failure cause outside program control)



MOV Functional Failures Last Four Quarters

	2003 QTR1	2003 QTR2	2003 QTR3	2003 QTR4	Totals
MPFF	0	0	0	0	0
Direct FF	0	3	0	0	3
Indirect FF	0	0	0	0	0

Failure Description	When
MO 2-1301-60 found with cracked stem nut	4/11/2003
MO 1-1001-43A found with torque switch roll pin broken	5/20/2003
MO 2-1301-16 found with pressure seal ring leaking	6/25/2003

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Engineering Based Valve Testing and Evaluation

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Abstract

Valve engineering and testing has a long history not only within FANP Germany (former Siemens KWU). The Siemens engineers began to develop and apply diagnosis-measurement equipment for valves as early as the 1980s. Initially, this equipment was designed for valve diagnosis measurement directly at valve locations. Evaluation of the results was based on the experiences of the engineers. We began to systemize the valve diagnosis and to link it to valve engineering in the 1990s. The Valve Performance Concept was developed. It represented the link between valve calculation, design evaluation, valve diagnosis and condition-oriented maintenance. The evaluation criteria of the diagnosis measurements were defined on the basis of the functional model of the valves and the allowable parameters were derived from valve calculation. In order to avoid the costly and time-consuming instrumentation and measurement of the valves in-situ, engineering-based evaluation methods as well as measuring equipment have been developed to determine all necessary diagnosis parameters based on active power measurement from the switch-gear. This idea resulted in our evaluation software ADAM® qualified by the authorities and several types of diagnosis equipment, e.g. SIPLUG®. Due to the active power measurement combined with the quantitative evaluation of the main features, deviations from the design tolerance levels can be identified in the whole chain from the power supply system to instrumentation and control (I & C), actuator and valve. This diagnosis and evaluation methodology is used today in many NPPs, mainly in western and eastern Europe. It is also applicable for testing according to U. S. NRC Generic Letter 96-05. The present FANP diagnosis measurement equipment is the Ultra Check family for measurement at valve locations and the SIPLUG® family for diagnosis based on active power measurement. The measurement equipment can be combined with the evaluation software ADAM®. Existing diagnosis measurement equipment and measurement results can be included as well. It allows the determination of the state of the valves anytime considering statistical evaluation and trending. The reduction of costs for diagnosis measurement and evaluation is possible. The concept of permanent monitoring with SIPLUG® online and ADAM® will be put into effect in the new NPP Olkiluoto 3 in Finland from the start. The results of permanent monitoring, trending and statistical evaluation will be considered for the planning of the scope of maintenance during outages.

Based on this concept, predictive maintenance planning of the outages is possible resulting in high reliability of the nuclear power plants (NPPs).

1. Introduction

Valve engineering and testing has a long history not only within FANP Germany (former Siemens KWU). Our valve engineers have been involved in the definition of requirements for nuclear valves and in the development of such valves since the beginning of nuclear technology in Germany. During the last 25 years, engineering work to a large degree focused on the development of valve diagnosis methods, equipment and evaluation. The application of valve diagnosis is one reason for the high reliability of valves in Siemens NPPs worldwide, represented by the high reliability of these NPPs. Return of investment was possible due to a justified change of maintenance practice from preventive to predictive maintenance. This presentation describes the development of the engineering-based valve diagnosis and evaluation from the beginning up to now considering, for example, valves with electrical actuators.

2. First Steps

The Siemens engineers began to develop and apply diagnosis-measurement equipment for valves as early as the 1980s. The intention was to implement a complete system of motor-operated valve (MOV) diagnosis equipment that allowed the verification of correct operation of the valves and the detection of potential deviations and faults. This system was meant to be applied for diagnosis during outages as well as during commissioning of NPPs. Initially, this equipment was designed for valve diagnosis measurement directly at valve locations. Diagnosis parameters were mechanical parameters like torque, stem thrust and actuator worm gear displacements as well as electrical parameters like switch signals and active power. The evaluation of the results was based on the experience of the engineers. There was no direct link between diagnosis and calculation/engineering although calculation results were considered. The evaluation included, e.g., the correct adjustment of the actuators (switch-off variant and torque switch settings) and checking the start-up torque (especially for globe valves).

3. Engineering based evaluation of diagnosis results

We began to systematize the valve diagnosis and to link it to valve engineering in the 1990s. The Valve Performance Concept was developed. It represented the link between valve calculation, design evaluation, valve diagnosis and condition-oriented maintenance. The evaluation criteria of the diagnosis measurements were defined on the basis of the functional model of the valves and the allowable parameters were derived from valve calculation.

From the beginning, valve calculation included the following steps:

- Verification of the required stem thrust and torque
- Selection of actuator
- Determination of maximum thrust and torque
- Strength analysis of parts in the load path to verify the capability of function
- Analysis of switch-off failure
- Stress and fatigue analysis of pressure retaining parts.

Variable parameters, like friction coefficients or switch-off tolerances, were considered within the verification of the required stem thrust and torque. Allowable ranges of these parameters were defined and covered by safety margins. The calculation methodology as well as the allowable ranges of the parameters and the applicable safety margins have been discussed and agreed with German authorities and are written down in calculation guidelines or German regulations like KTA guidelines. Special computer software is available for calculations according to these guidelines.

In order to avoid the costly and time-consuming instrumentation and measurement of the valves in-situ, engineering-based evaluation methods as well as measuring equipment have been developed to determine all necessary diagnosis parameters based on active power measurement from the switch-gear. This idea was resulted in our evaluation software ADAM® qualified by the authorities and several diagnosis equipment, e.g. SIPLUG®. The evaluation software ADAM® includes project-specific databases with the evaluation criteria for all diagnosis-relevant valves. These evaluation criteria are derived from the valve calculation considering relevant safety margins.

The following parameters (minimum and maximum values) are used as evaluation criteria:

- Start-up torque
- Running torque
- Switch-off torque
- Final torque
- Torque rate (start-up and end position)
- Stroke time
- Switch-off delay
- Friction coefficient

The measurement equipment based on active power measurement allows the recording of the active power and the determination of the following parameters considering the calibration curves of the actuator:

- Start-up torque
- Running torque
- Switch-off torque
- Torque rate (start-up and end position)
- Stroke time
- Tightening time (end position)
- Switch-off delay
- As derived parameter: Friction coefficient

Our evaluation software ADAM® is used to determine the characteristic parameters of the diagnosis measurement (see above). The stem factor is determined based on the in/out-factor and run-time-method. The acceptability of the determined parameters is evaluated by comparison with the allowable values given in the ADAM®-database. The accuracy of the measurement and resulting calculations is taken into account during the comparison. After the evaluation (***Figure 1***), the measurements are displayed in a list (***Figure 2***). Each line in the list shows information regarding one measurement. This list contains the MOV's tag number, date and time of the measurement and an overall assessment ("OK", "uncertain" or "fault detected"). Red colored arrows and frames indicate that a parameter is below or above the given limits. Blue checkmarks indicate correct results. All measurements can be graphically displayed. The measurement results can be used for statistical evaluation and trending. Trending shows long-term changes of relevant parameters displaying them across time. The statistic function displays selected parameters for multiple MOVs. In addition, the reference values and limit values are shown.

The evaluation of the diagnosis measurement based on these data allows the detection of most of the potential faults noted in U.S. NRC Generic Letter 89-10:

- Incorrect torque switch setting
- Spring pack gap or incorrect spring pack preload
- Incorrect stem packing tightness
- Excessive inertia
- Loose or tight stem-nut locknut
- Incorrect limit switch settings
- Stem wear (in the thread)
- Bent or broken stem
- Worn or broken gears
- Grease problems
- Motor insulation or broken rotor rods (2)
- Incorrect wire size or degraded wiring (2)
- Disk/seat binding (including thermal binding)
- Motor undersized (1)
- Mal-adjustment for failure of hand wheel declutch mechanism
- Relay problems
- Worn or broken bearings
- Broken or cracked limit switch and torque switch components
- Missing or modified torque switch limiter plate
- Hydraulic lockup
- Degraded voltage (within design basis)
- Defective motor control logic (1)
- Excessive seating or back-seating force application
- Incorrect reassembly or adjustment after maintenance (1)
- Unauthorized modification or adjustments (1)
- Torque switch or limit switch binding

(1) faults that can be detected under some circumstances but not in all cases

(2) by current measurement and current symmetry

In addition to the potential faults listed above, other common failures can be identified:

- Improper stroke times or improper stroke sequence times
- Excessive torques and stem thrusts
- Overstrain of valve parts in the load path
- Loss of self-locking of the stem nut
- Loss of self-locking of the actuator worm shaft
- Wear or defects on the stem nut bearings
- Improper design or assembling of disc springs for stem nut support
- Increase or decrease of actuator efficiency
- Increase or decrease of stem nut friction coefficient
- Faulty contactors (main contactors)
- Unsteady behavior during valve run (fluctuation of running power)

Due to the active power measurement combined with the quantitative evaluation of the main features, deviations from the design tolerance levels can be identified in the whole chain from the power supply system to I & C, actuator and valve. The evaluation criteria for the databases can be calculated before the start of the first diagnosis and can be used for all steps of diagnosis: Factory Acceptance Tests at the valve manufacturer, commissioning of valves, diagnosis during outages or during operation.

Considerable commercial effects can be achieved with this diagnosis measurement and evaluation by ADAM®. The measurements and evaluations can take place completely self-controlled during plant operation. The condition of the valves can be checked in advance before the outages. Statistic and trending allow extrapolation of the valve conditions into the future. Critical valves can be detected and evaluated in more detail and/or monitored permanently. Valves identified for maintenance and justified by engineering can be taken into account for the outage planning. Thus, the scope and duration of valve inspection/maintenance during outages can be optimized. Unnecessary maintenance activities can be avoided.

Evaluation is used today in many NPPs, mainly in western and eastern Europe. The diagnosis methodology is also applicable for testing according to U.S. NRC Generic Letter 96-05.

4. Present diagnosis equipment

The present FANP diagnosis measurement equipment is the Ultra Check family for measurement at valve locations and the SIPLUG® family for diagnosis based on active power measurement.

As an example the three versions of SIPLUG® are described below:

- Diagnosis sockets with external SIPLUG®
- Pocket SIPLUG®
- SIPLUG® online

Diagnosis sockets with external SIPLUG® (Figure 3)

For measurement of active power, 2 or 3 inductive current transformers and a diagnosis socket are permanently installed in the switch gear. The current transformers can be mounted in the cable outlet area or inside the plug-in unit. The current transformers are easy to install - the power wires of the three phases are fed through the holes of the transformers.

The diagnosis socket can be mounted on the front panel of the plug-in units or in the back doors of the cabinets. For safety reasons, the connections between the diagnosis socket and the power circuit are protected by fuses.

SIPLUG® is a low-cost, battery-powered, miniature data acquisition and storage device.

When the valve is operated, the voltages and currents are measured. The active power is then calculated from these measurements and stored in the SIPLUG®'s internal memory. A total of 400 seconds of data can be stored in the SIPLUG® memory. If the memory is full, the oldest measurements are replaced by the new ones. SIPLUG® measurements can be read directly by the ADAM® software and stored on hard disk. The connection to the computer is made via the standard serial port.

For a measurement, a SIPLUG® is plugged into the diagnosis socket (***Figure 4***). It continuously monitors the control voltages of the interface relay. If a control voltage is detected, data acquisition and storage will occur until the control voltage drops and the motor voltage is zero.

Each diagnosis socket contains a unique code that can be read by the SIPLUG®. From the socket code, the SIPLUG® can determine which MOV is being measured. Furthermore, the user does not need to select an MOV identifier for storing the data - the ADAM® evaluation software automatically performs all data handling via the socket code including the automatic selection of the power range. One SIPLUG® can record data from different MOVs.

Pocket SIPLUG® (Figure 5)

The Pocket SIPLUG® was developed to allow an adequate measurement from switch gears which are not equipped with diagnosis sockets and installed current transformers. The Pocket SIPLUG® is directly adapted to the switch gear by current clamps. The diagnosis functions are similar to the diagnosis socket/external SIPLUG®.

Advantage of this solution: It can also be applied for diagnosis measurement from the valve actuator because the Pocket SIPLUG® can be adapted as well directly to the actuator. The recording and evaluation of data can be completed by mechanical parameters like torque and/or thrust. Existing diagnosis measurement equipment and measurement results can be included as well.

The Pocket SIPLUG® is the simplest start of this diagnosis technology and does not require any modification of the switch gear.

SIPLUG® online (Figures 6 and 7)

The latest development of the valve diagnosis is an online method with automatic engineering-based evaluation, although other applications are still in use.

Small SIPLUG®-online measurement modules are the basis for this variant. They are permanently installed in the switch-gear and allow an automatic active power measurement. These SIPLUG®-online modules are qualified and calibrated measurement equipment. Each valve operation is measured, saved and evaluated for all accordingly equipped valves. The measured data are sent via a data-bus to a central diagnosis server and saved there.

The evaluation software ADAM® is identical for all three SIPLUG® versions. It is also possible to have a combination of the three versions in one plant.

5. Present application of the ADAM®/ SIPLUG® concept

The concept of permanent monitoring will be put into effect in the new NPPs Olkiluoto 3 in Finland and the EPR in France from the start. All safety-related valves will be equipped with the SIPLUG®-online modules. The diagnosis methodology will be used first during the factory acceptance tests at the manufacturer, during commissioning, and later on during operation and outages to reduce preventive maintenance. The results of the permanent monitoring, trending and statistical evaluation will be considered for the planning of the scope of maintenance during outages.

This monitoring concept has influence on the complete valve engineering work:

- The valve specifications contain requirements for valve monitoring up to valve commissioning.
- The valve manufacturer has to present a valve calculation which allows the determination of diagnosis evaluation criteria. The manufacturer has also to specify the variable parameters and their allowable ranges.
- The valve actuators will be calibrated during the Factory Acceptance Tests (FAT).
- The variable parameters (e.g., friction coefficients) will be verified during the FAT of the valves. The measurement will be performed with measurement equipment adequate to the on-site monitoring. The evaluation of the results will consider the specified evaluation criteria. The FAT is the basis measurement for the on-site monitoring.
- The commissioning of the valves in the plant will be used as basic on-site monitoring measurement.

This monitoring concept enables us to improve an item which in the past could not be covered satisfactorily by our engineering concept:

Very low friction coefficients for stem/stem nut were detected in different globe valves with higher stem diameters. These very low friction coefficients <0.05 resulted in the loss of self-locking and self-opening of the valves because of a non self-locking transmission gear of the actuator. In addition, very high stem thrust was induced with high stresses in valve parts.

The stem nut was replaced in case of low friction coefficients in the past to keep the friction coefficient within the allowable range required by the German calculation guidelines.

In the future, we will accept valve calculations with small friction coefficients. The valve manufacturer must define the allowable range and consider it in the calculation. The acceptability of the actual friction coefficient will be checked during FAT and periodically monitored on-site. The loss of self-locking must be avoided by design features, e.g. by using self-locking actuators.

6. Summary

The presentation shows that a simple and permanent monitoring of valves in NPPs is possible with the presently available diagnosis equipment and methodology as well as engineering-based evaluation methods. Existing diagnosis measurement equipment and measurement results can be included as well. The reduction of costs for diagnosis measurement and evaluation is possible (***Figure 8***). It allows anytime the determination of the state of the valves considering statistical evaluation and trending. Based on this concept, a predictive maintenance planning of the outages is possible resulting in high reliability of the NPPs. However, this has to be accompanied with a reliable engineering work based on a qualified performance prediction methodology, e.g., as justified in the U.S. by the Electric Power Research Institute (EPRI). In addition, FANP has also engineering-based diagnosis methods and equipment for pilot operated valves, air operated valves and solenoid operated valves.

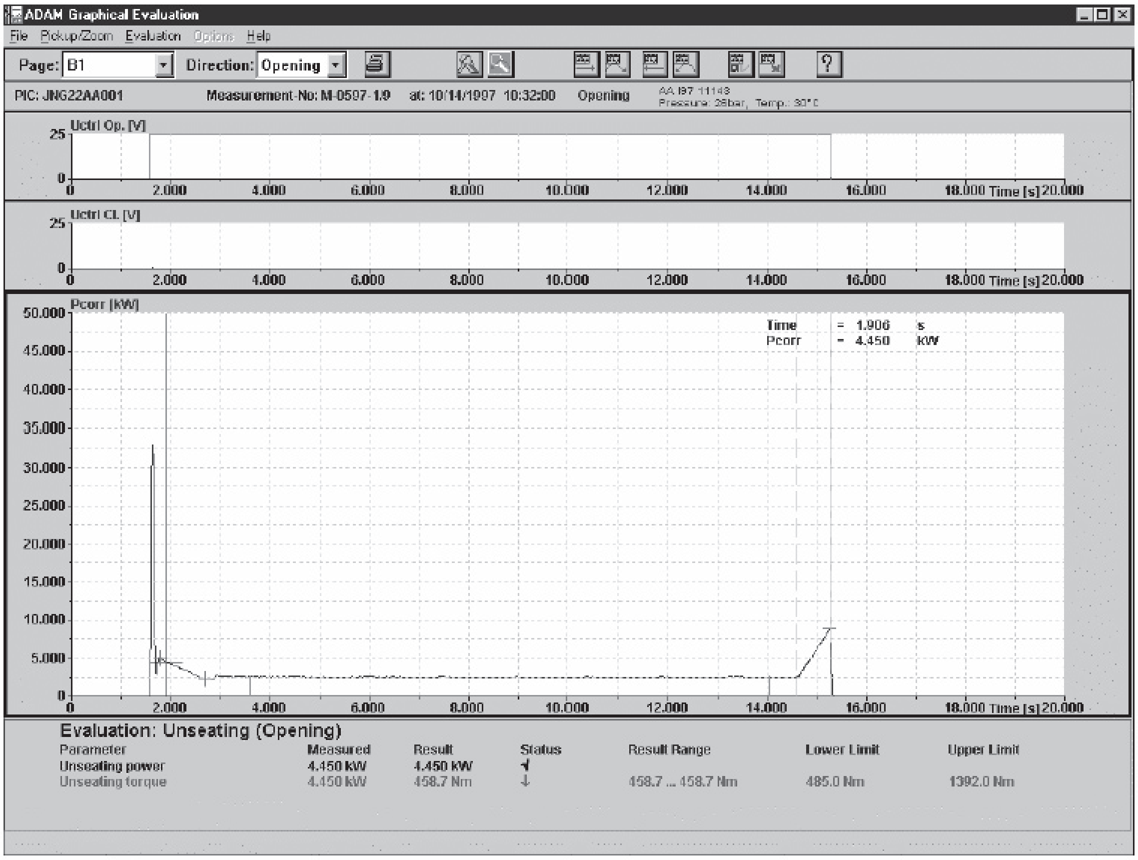
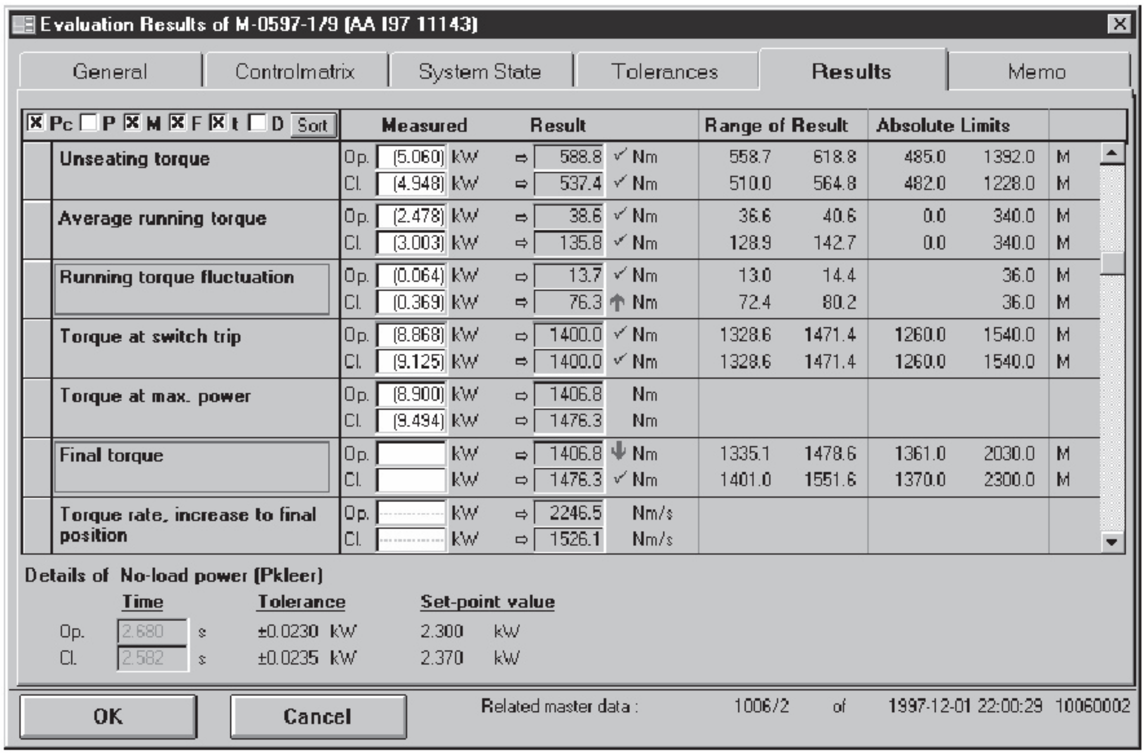


Figure 1: **Diagnosis evaluation with ADAM ® – Active power diagram**


Evaluation Results of M-0597-1/9 (AA I97 11143)

General | Controlmatrix | System State | Tolerances | Results | Memo

Pc P M F t D		Measured		Result		Range of Result		Absolute Limits		
Unseating torque	Op.	(5.060)	kW	588.8	Nm	558.7	618.8	485.0	1392.0	M
	Cl.	(4.948)	kW	537.4	Nm	510.0	564.8	482.0	1228.0	M
Average running torque	Op.	(2.478)	kW	38.6	Nm	36.6	40.6	0.0	340.0	M
	Cl.	(3.003)	kW	135.8	Nm	128.9	142.7	0.0	340.0	M
Running torque fluctuation	Op.	(0.064)	kW	13.7	Nm	13.0	14.4		36.0	M
	Cl.	(0.369)	kW	76.3	Nm	72.4	80.2		36.0	M
Torque at switch trip	Op.	(8.868)	kW	1400.0	Nm	1328.6	1471.4	1260.0	1540.0	M
	Cl.	(9.125)	kW	1400.0	Nm	1328.6	1471.4	1260.0	1540.0	M
Torque at max. power	Op.	(8.900)	kW	1406.8	Nm					
	Cl.	(9.494)	kW	1476.3	Nm					
Final torque	Op.		kW	1406.8	Nm	1335.1	1478.6	1361.0	2030.0	M
	Cl.		kW	1476.3	Nm	1401.0	1551.6	1370.0	2300.0	M
Torque rate, increase to final position	Op.		kW	2246.5	Nm/s					
	Cl.		kW	1526.1	Nm/s					

Details of No-load power (Pkleer)

	Time		Tolerance	Set-point value
Op.	2.680	s	±0.0230 kW	2.300 kW
Cl.	2.582	s	±0.0235 kW	2.370 kW

OK Cancel Related master data : 1006/2 of 1997-12-01 22:00:29 10060002

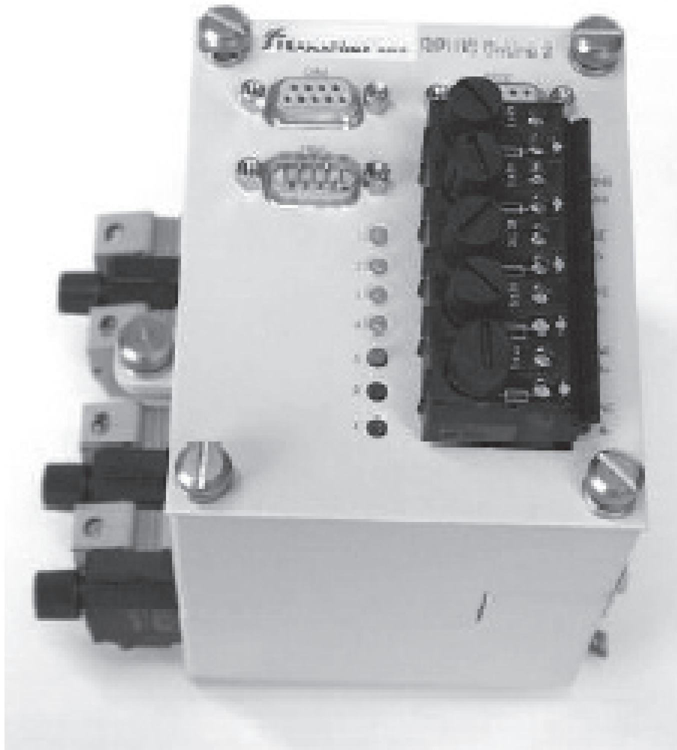
Figure 2: **Diagnosis evaluation with ADAM ® – Result list of evaluation parameters**

Figure 3: **External SIPLUG®**

Figure 4: **Switchgear equipped current transformers inside the plug-in unit and with diagnosis sockets for adaptation of the external SIPLUG®**

Figure 5: Pocket SIPLUG® with current clamps and transportation case

Figure 6: SIPLUG® online 2 module for installation in the cable outlet



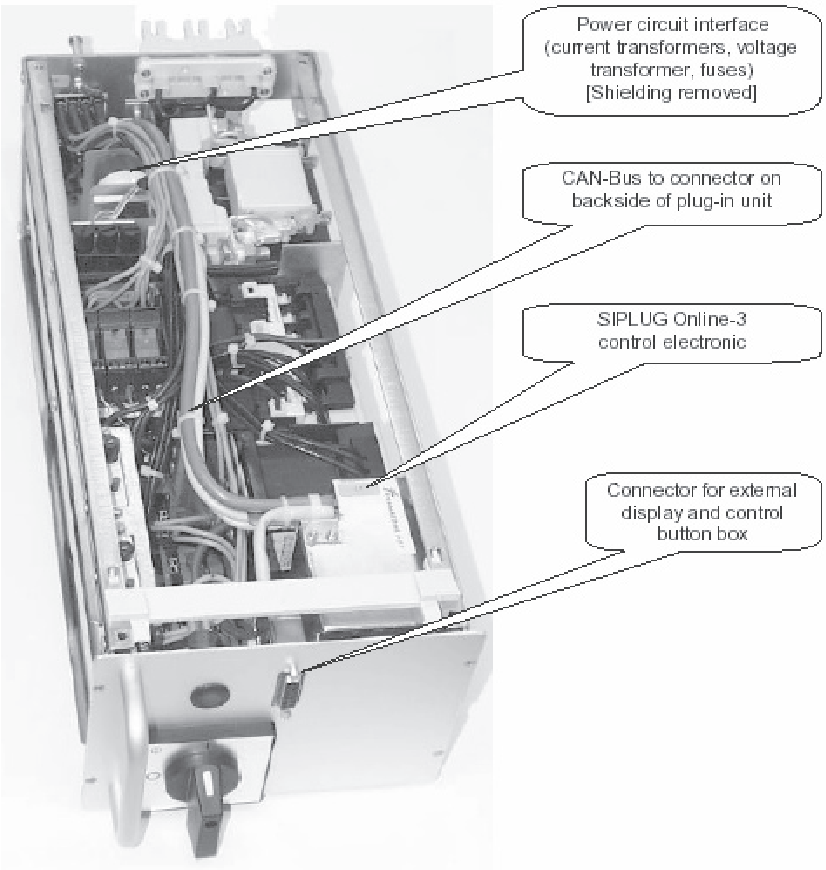


Figure 7: **SIPLUG® online 3 module (integrated in switch gear plug-in module)**

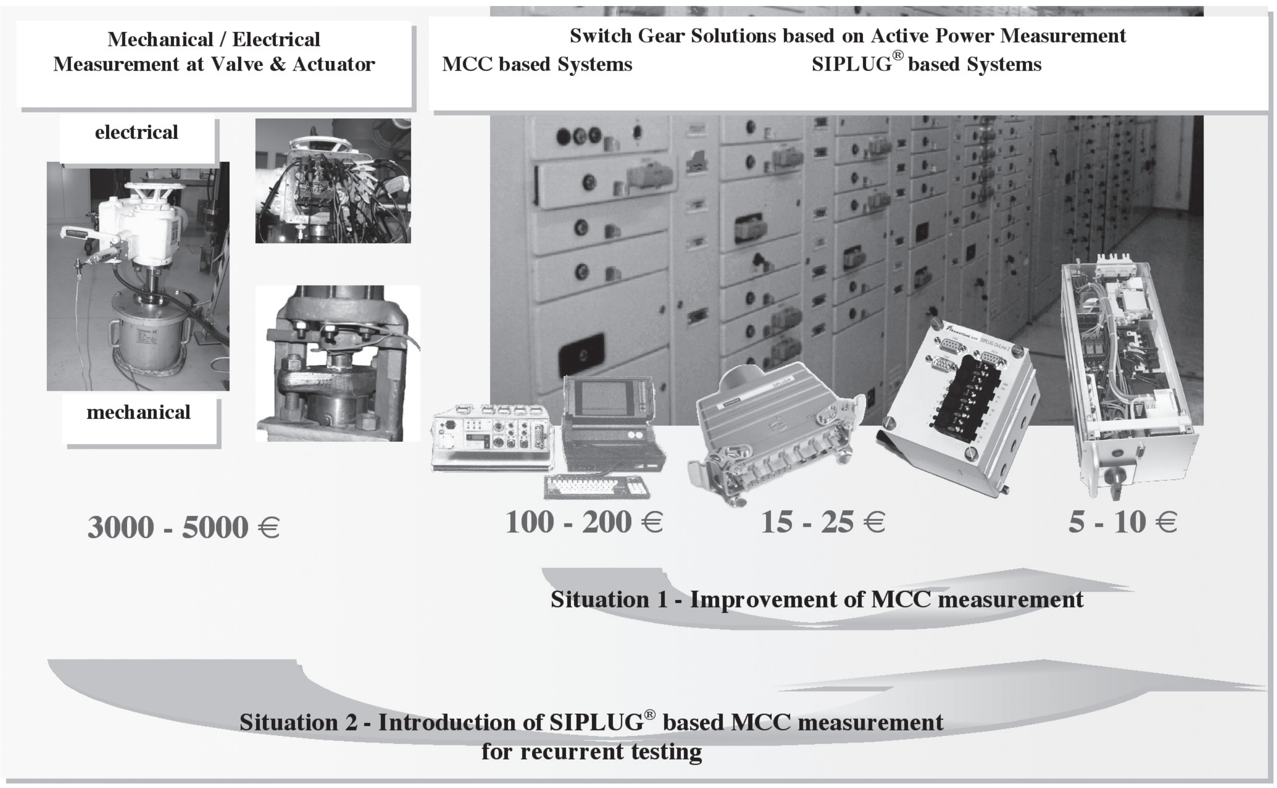


Figure 8: **Recurrent testing and estimated costs**

MOV Periodic Verification Approach from the Joint Owners' Group Program

Tim Chan
TVA

Robert Doyle
Arizona Public Service

Chad Smith
Duke Energy

Glenn Warren
Southern Nuclear Operating Co.

Paul Damerell
Todd Spears
MPR Associates

Abstract

To address long-term motor operated valve (MOV) performance, the Babcock & Wilcox, Boiling Water Reactor and Westinghouse Owners' Groups conducted the Joint Owners' Group MOV Periodic Verification (PV) Program. This program, now complete, had participation by 98 of the 103 operating U.S. reactor units. The program provides a justified approach for periodically testing MOVs. The technical basis is a series of repeat tests on 176 gate, butterfly and globe valves, performed at the participating plants. The PV approach classifies each valve and then specifies a PV test interval based on the MOV's margin and risk significance.

The in-plant repeat testing was performed under conditions with flow and differential pressure (DP) in the pipe. Valves were tested three times, with at least a year between tests. The test results show that there was no age-related degradation, i.e., no increases in required thrust or torque simply due to the passage of time, without DP stroking.

For gate valves, the required thrust did not degrade in service except under certain conditions. Specifically, when the initial valve factor is low due to either valve disassembly or due to limited DP stroking in service, the valve factor tends to increase with DP stroking, up to a stable level. To address this observation, the gate valve PV method includes threshold values above which increases are not observed. Because different valves stabilize at different valve factors, the PV method also provides ways for users to demonstrate from testing that the required thrust is stable.

For butterfly valves, the required torque did not degrade in service, but certain bearing materials and fluid conditions showed variations in bearing friction coefficient, even though there was no increasing or decreasing trend. To address this observation, the butterfly valve PV method includes maximum bearing friction coefficients, as well as test-based methods for users to demonstrate that their friction is less than the maximum value.

For globe valves, no degradation in required thrust was observed, and no limits or test methods are included in the globe valve PV method.

Keywords: periodic verification motor operated valve degradation

Background

US nuclear power plants expended significant efforts in the 1990s to improve MOV reliability and to satisfy US Nuclear Regulatory Commission (NRC) Generic Letter (GL) 89-10 (Reference 1). Periodic verification of MOVs is separately covered in NRC GL 96-05 (Reference 2).

To address GL 96-05, the nuclear industry sought to take advantage of the investments each plant made in their GL 89-10 programs and of subsequent testing. The Joint Owners' Group (JOG) MOV Periodic Verification (PV) Program was formed on this basis. Specifically, the Babcock & Wilcox Owners' Group (B&WOG), Boiling Water Reactor Owners' Group (BWROG), Combustion Engineering Owners' Group (CEOG) and Westinghouse Owners' Group (WOG) joined together for the JOG MOV PV Program. During the program, the CEOG merged into the WOG.

The objective of the JOG MOV PV Program is to provide an approach for MOV periodic verification. At the outset of the JOG MOV PV Program (1997), a Program Description Topical Report was prepared (Reference 3). This report described the "design" of the program and the underlying technical basis. This report was submitted to the NRC, who subsequently issued a Safety Evaluation (Reference 4) accepting the proposed program. Individual plants notified the NRC whether they were participants in the JOG MOV PV Program or whether they were implementing their own approach for periodic verification. Ninety-eight (98) of the 103 operating reactor units in the US participated in the JOG MOV PV Program.

This united approach used in the JOG MOV PV Program has key benefits for participating plants and for the regulator. Importantly, it conserves resources. Cost effectiveness is achieved by sharing the burden of valve testing among participating plants. Also, because the program provides a uniform approach for all participating plants, the regulator's burden to individually inspect and approve multiple programs is alleviated. Accordingly, plants can operate under a predictable regulatory expectation with high certainty of acceptance. Finally, because the program has 98 participating units, an extensive set of MOV test data was obtained and evaluated. These data, which are far more extensive than any single plant could expect to obtain, provide the basis for a strong technical justification.

The scope of the JOG MOV PV Program covers the potential degradation in required thrust or torque. The JOG MOV PV program does not cover potential degradation in actuator available thrust or torque. This element of potential degradation is the responsibility of each individual plant, and the JOG MOV PV approach identifies where this degradation should be considered.

In-Plant DP Testing

As mentioned above, a key element of the JOG MOV PV Program is MOV testing at the participating plants. Each participating unit tested two valves under conditions with flow and differential pressure (DP). Each valve was tested three times under nominally identical DP conditions, with at least a one-year separation between tests. The test valves were selected so that, in aggregate, they cover the valve design features and system conditions most commonly encountered in nuclear power plants.

The DP test program includes 176 valves: 134 gate valves, 23 butterfly valves, 12 unbalanced disk globe valves, and 7 balanced disk globe valves. Data were obtained from 3 tests of each valve for 161 of the valves; the remaining 15 valves yielded data for only 2 tests. In total, data from 513 tests were obtained.

To ensure that data obtained from in-plant tests were satisfactory for use in the JOG MOV PV Program, the participating plants were required to adhere to a test specification (included in Reference 3), which includes requirements for:

- Test valve maintenance and material condition, both before and during the tests
- Test conditions
- Test instrumentation
- Test sequence
- Test data evaluation
- Test documentation

The goal of the standard test specification was to ensure that all valves and testing were properly controlled to achieve adequate consistency and quality in the test results obtained from multiple plants. Importantly, the test specification requires that time-history data for stem thrust (or torque for butterfly valves) and DP be obtained. Further, the specification requires analyzing and summarizing the data in a prescribed manner. Finally, the specification requires a test sequence that includes both static and DP test strokes. Although there was not a minimum permissible DP, the specification required that the DP be closely repeated between tests.

Program Completion and Key Conclusions

Four previous papers (References 5, 6, 7 and 8) describe the JOG MOV PV Program and show interim results from in-plant valve tests. The testing is now complete. The purpose of this paper is to summarize the tests results and the insights gained in the program, and to describe the recommended periodic verification approach. A new topical report describing the test results and the PV approach has been prepared and submitted to the NRC (Reference 9). At the time of this paper, the NRC was performing their review.

The key conclusions from the test results are as follows.

- There is no age-related degradation for gate, globe and butterfly valves, i.e., no increase in required DP thrust or torque only due to the passage of time (without DP stroking).
- For gate valves, service-related degradation (increase in required thrust with DP stroking) occurs only with valves that have a low initial valve factor due to disassembly/reassembly or due to limited DP stroking in service. In these cases, the valve factor tends to increase with DP stroking, up to a stable level.
- For butterfly valves, there is no service-related degradation. Butterfly valves with bronze or 300 series stainless steel bearings in untreated water systems without hub seals show variations in bearing friction, with no increasing or decreasing trend. Valves with non-metallic bearings also show small variations.
- For balanced and unbalanced disk globe valves, there is no service-related degradation. Balanced disk globe valves is untreated water systems show thrust variations unrelated to DP thrust. These variations have no increasing or decreasing trend and appear to be related to the effect of particulates.

Overall Periodic Verification Approach

Based on the evaluation of the data, a recommended periodic verification approach has been developed. The JOG MOV periodic verification approach is to classify each applicable valve into one of four classes. The periodic verification requirements are defined for each class based on the valve's risk ranking and margin. Because this PV approach addresses the potential degradation in required thrust or torque, appropriate allowances for actuator degradation need to be included in the calculation of margin. The four classes are summarized below.

Class A

Class A valves are not susceptible to degradation, as supported directly by testing performed in the JOG MOV PV Program. For these valves, static PV testing is only needed to verify proper MOV setup and to quantify margin. For Class A valves with positive margin, the interval between static PV tests is based on the "High Margin" column of Table 1: six years for high risk valves and ten years for medium and low risk valves. The justification is that, because there is no susceptibility to degradation in required thrust, the longest interval is acceptable.

Class B

Class B valves are not susceptible to degradation based on the test results in the JOG MOV PV Program, extended by analysis and engineering judgment to configurations and conditions beyond those tested. For these valves, static PV testing is only needed to verify proper MOV setup and to quantify margin. For Class B valves, the interval for static PV testing is determined from Table 1. The justification is that Class B valves are not susceptible to degradation in required thrust, but the certainty is not as high as for Class A. Therefore, full use of the table, rather than just the high margin column, balances the decreased certainty.

Class C

Class C valves are susceptible to changes in required thrust or torque, as shown by test results in the JOG MOV PV Program. Potential increases in required thrust or torque need to be taken into account in the setup, surveillance and evaluation of these valves. For Class C valves, the PV requirements tend to force changes in the valve or its setup so that it can be reclassified as Class A or B. For gate valves, an allowance needs to be considered in computing the valve's margin. If the margin (including allowance) is positive, static PV testing in accordance with the intervals in Table 7-1 is to be used. For all butterfly valves and for gate valves where the margin (including allowance) is forecast to be less than zero, either (a) the valve is to be DP tested (rather than static tested) at a 2 year interval, with the first DP test to occur at the next available opportunity, not to exceed 2 years, or (b) the MOV or its setup is to be modified such that it covers potential increases or variations in required thrust or torque. Note that globe valves cannot be Class C.

Class D

Valves in Class D are not covered by the JOG MOV PV Program. Individual plants are responsible for justifying the PV approaches for these valves. Valves that are classified as Class D tend to be valves that have a combination of specific, unusual design features in conjunction with certain application conditions. For example, gate vales with self-mated 300 series stainless steel guides that stroke in service above 120ºF are Class D, and globe valves with rising/rotating stems that stroke open against DP are Class D. These specific configurations and applications have potential degradation mechanisms not covered by the JOG MOV PV Program testing.

Periodic Verification of Gate Valves

Figure 1 shows a typical gate valve. The stem moves a wedge-shaped disk into or out of the flow stream to close or open the valve. The required thrust to move the disk needs to overcome packing friction, the effect of pressure pushing the stem out of the valve (stem rejection) and friction of internal valve surfaces sliding against each other. Only the last term is affected by the presence of flow and DP across the valve during its stroke.

The gate valve test data from the JOG MOV PV Program are extensive, and they were analyzed in several ways to evaluate potential degradation in required thrust. These evaluations showed that disk-to-seat friction is the dominant influence on required thrust, and that periodic verification needs to consider circumstances where this friction could increase above the value currently used to justify valve setup and to quantify margin.

Gate valve test data were analyzed to isolate disk-to-seat friction by examining the portions of closing and opening strokes where the disk is sliding across the seat ring. This sliding occurs toward the end of closing strokes (after the disk has covered the seat ring but before it wedges) and at the beginning of opening strokes (after unwedging but before a flow passage opens). The apparent disk-to-seat friction (expressed as either a "valve factor" or a friction coefficient) can be determined from measurements of thrust, line pressure and differential pressure. The results from repeat tests conducted over a span of a few years can then be evaluated to determine the trend. Figure 2 shows typical results. This graph shows the mean and range of disk-to-seat friction (expressed as a valve factor) for a group of 27 valves tested in cold (<120ºF), treated water. These valves have Stellite disk and seat faces and are in service where they stroke against DP 1 to 4 times per year. The results are subdivided into 2 categories – valves that were disassembled and reassembled prior to (within two years of) the first test, and valves that were not disassembled. The disassembled valves exhibit lower initial valve factors that tend to increase in subsequent tests up to a level similar to non-disassembled valves. The DP stroking appears to be responsible for the increase. Figure 3 shows average valve factors for valves (both disassembled and non-disassembled) in 3 categories: valves not typically DP stroked, valves DP stroked 1 to 4 times per year, and valves DP stroked more than 4 times per year. Valves that are DP-stroked more often show a larger, more rapid rise than those that were stroked less frequently.

Another key observation was that different gate valves tend to stabilize at different valve factors; hence, there is a range of potential stable valve factors. If a valve currently has a valve factor in the lower part of the range, it might be susceptible to increase or it might be stable. Valves that had low valve factors and that do not typically DP stroke in service were the most susceptible to increases.

Similar results were observed for gate valves in other fluids (e.g., hot treated water, untreated water, steam) and for valves with other disk-to-seat materials. Figure 4 shows results for a set of eight valves in steam service. These valves all had Stellite disk-to-seat faces. For these valves, the effects of disassembly and stroking appear to be less than in cold treated water. Figure 5 shows results for a set of 4 valves with 400 series stainless steel disk faces and Stellite seat ring faces. The effect of disassembly can be clearly seen on one valve tested in water. Another disassembled valve in water shows minimal effect, because this valve was stroked multiple times between the disassembly and the first test. The steam valve shows minimal effect of disassembly.

Additional evaluations of the gate valve data were performed to evaluate disk guide-to-body guide friction and the friction between the parts of multi-piece disks. These evaluations tended to show stable friction. The effects of disassembly could be seen in the guide friction evaluations, but these effects were less than those for disk-to-seat friction. Figure 6 shows guide friction results for 4 valves with Stellite disk guide faces and carbon steel body guide faces. One of these valves was disassembled, and the friction is stable for all 4 valves. Figure 7 shows results for 10 valves with 300 series stainless steel disk guide faces and either 300 series or 17-4 PH stainless steel body guide faces. Some friction increases can be seen in the valves that were disassembled; overall the results are stable.

The observed results for gate valves suggest that the potential for required thrust to increase depends on the current value of disk-to-seat friction coefficient used for valve setup and margin calculation, and its basis. A valve that has been shown by test to be stable at a specific friction coefficient will not show future increases. A valve that has not been shown by test to have a stable friction coefficient might be susceptible to future increases, depending on the current value. Figure 8 shows a plot of the change in friction coefficient (between consecutive JOG tests separated by at least a year), plotted against the initial friction coefficient. Values at the high end of the range tend be stable, but lower values are susceptible to increase. Based on this result, a periodic verification classification approach that considers the basis for disk-to-seat friction was developed.

First, a screen is used to determine which valve applications are covered by the test data, which are covered by extension and which are not covered. The screen considers: disk style, extent of in-service DP stroking, disk-to-seat and disk guide-to-body guide materials, fluid type, and stroke direction for the valve's design basis function. For valves that are either covered or covered by extension, two questions are evaluated. First, does that valve have a "qualifying basis" of test data that demonstrates that the value of disk-to-seat friction coefficient is stable? Second, does the disk-to-seat friction coefficient exceed the "threshold" value that characterizes a 95% non-exceedence level, as supported by the JOG MOV PV Program test data? A "yes" answer to either of these questions means that the basis for the required thrust for the valve is reliably stable, and the valve is classified as Class A or B, as appropriate. If the answer to both questions is "no", then the valve is susceptible to increases in DP thrust and the valve is classified as Class C. Figure 9 shows a flow chart of the classification process.

Periodic Verification of Butterfly Valves

Figure 10 shows a typical butterfly valve. The stem turns a disk, typically through a 90º stroke. In the closed position, the disk mates with a seat ring on the body inner diameter and blocks the flow. In the open position the disk is parallel to the flow stream, allowing significant open area for flow. The required torque to move the disk needs to overcome packing friction, disk-to-seat friction (only near the fully closed position), stem bearing friction and hydrodynamic loads applied to the disk by the flow. Only the last two terms are affected by the presence of flow and DP across the valve during its stroke. Further, the hydrodynamic load term is not susceptible to degradation. Accordingly, the JOG MOV PV Program examined only the bearing friction term.

Butterfly valve bearing friction was determined from test data by comparing the valve's performance, near the fully closed position, under conditions with and without DP. Because the hydrodynamic torque is negligible in this part of the stroke, the difference in required torque is entirely due to bearing friction. Measurements of stem torque and DP, along with the known diameters of the stem and disk, are sufficient to determine the stem-to-bearing friction coefficient.

Figure 11 shows the bearing friction coefficients for 4 butterfly valves with bronze bearings, in applications with treated water < 100ºF flowing in the pipe. (Values are not shown on the y-axis because they are not needed to understand the observed trend.) Results are shown for the baseline, second and third tests (two strokes per test). There is more than one year of separation between tests. The bearing friction is observed to be stable and there is no increasing trend. One valve showed a significant decrease from the baseline to the second test; a careful review of the data showed that this observation was due to an unusually low unseating torque measured in the baseline static (no DP) test, and that the performance with DP was stable.

Figure 12 shows the bearing friction coefficients for 7 butterfly valves with bronze bearings, in applications with untreated water < 100ºF flowing in the pipe. The results are subdivided into two groups: 3 valves have bearing hub seals and demonstrate low, stable friction; 4 valves do not have bearing hub seals and demonstrate higher friction with considerable variations. The variations do not have an increasing or decreasing trend. Further, the changes are unrelated to the amount of DP stroking that the valve undergoes. Sometimes variations occur between consecutive strokes performed on the same day, in other cases the variations occur between stokes performed years apart. For these conditions (bronze bearing, untreated water, no hub seal), a single measured value of bearing friction cannot reliably be assumed to be stable.

Figure 13 shows results for Teflon-lined bearings in both treated and untreated water. The friction coefficient in untreated water tends to be a little higher, and show a little more variation, than in treated water. Overall, these results are lower than those for bronze bearings, and show less variation than bronze bearings in untreated water.

Figure 14 shows results for 4 valves with 4 other non-metallic bearing materials: Tefzel, polyethylene, Nomex and Nylatron. These results are relatively stable, although the very low friction coefficients for Nylatron in untreated water show some variation.

The observed results for butterfly valves indicate that some bearing materials and fluid conditions have stable bearing friction while other combinations have variations in bearing friction. For those valves that are susceptible to variation, either a set of tests is needed to establish a "qualifying basis" for bearing performance, or an appropriate "threshold" value of bearing friction coefficient (that covers the variations) needs to be used to set up the valve and determine its margin. Based on this result, a periodic verification classification approach was developed that considers bearing material and fluid conditions, the presence or absence of a hub seal, and for those conditions with variations, the basis for bearing friction coefficient.

First, a screen is used to determine which valve applications are covered by the test data, which are covered by extension and which are not covered. The screen considers: bearing and shaft materials, fluid type, and presence or absence of a hub seal. Valves that have bearing materials and fluid conditions not susceptible to variation are identified and classified as Class A. For valves that are susceptible to variation, two questions are evaluated. First, does that valve have a "qualifying basis" of test data that demonstrates that the value of bearing friction coefficient covers the variation? Second, does the bearing friction coefficient exceed the "threshold" value that characterizes a 95% non-exceedence level, as supported by the JOG MOV PV Program test data? A "yes" answer to either of these questions means that the basis for the required torque for the valve is reliable, and that the valve is classified as Class A or B, as appropriate. If the answer to both questions is "no", then the valve is susceptible to increases in DP thrust and the valve is classified as Class C. Figure 15 shows a flow chart of the classification process.

Periodic Verification of Balanced Disk Globe Valves

Figure 16 shows a typical balanced disk globe valve. The stem moves a disk toward or away from a seat to close or open the valve. A balancing port in the disk allows the pressures above and below the disk to be identical. A sliding seal at the end of the disk away from the seat separates the upstream and downstream pressures. Resistance to disk motion comes from packing and sliding seal friction, the effect of pressure pushing the stem out of the valve (stem rejection), area imbalance of the upper and lower sealing diameters on the disk, and friction between the disk and its internal guiding surface. Only the last two terms are affected by the presence of flow and DP across the valve during its stroke, and the area imbalance term is not susceptible to degradation. Accordingly, only a potential increase in disk-to-guide friction could produce a degradation (increase) in required DP thrust.

From the test data, the entire DP thrust (including imbalance and internal friction) was determined and expressed as a valve factor. The first observation from the data is that the DP thrust for these valves is very small, in most cases smaller than the packing friction. Therefore, these valves are inherently insensitive to degradation in required DP thrust. Further, the DP thrust was observed to be stable, i.e., no degradation was observed. Figure 17 shows the results for closing strokes of balanced disk globe valves, and Figure 18 shows the results for opening strokes. (Values are not shown on the y-axis because they are not needed to understand the observed trend.) These test results are from applications in water less than 120ºF and cover a variety of disk-to-guide materials. For both opening and closing, the average result is steady across three tests. Analysis of the data showed that the variations observed for individual valves are within the measurement uncertainty of the tests.

For 3 balanced disk globe valves tested in untreated water, thrust variations unrelated to DP were observed in some tests and not in other tests. These variations appeared as increases in thrust in certain portions of the stroke that had no buildup of DP. These increases were ascribed to the accumulation of particulate matter in the valve, and the plants found that periodically exercising the valve was effective in eliminating this effect.

Because balanced disk globe valves are insensitive to degradation and no degradation was observed, a periodic verification approach of periodic static testing (Class A or B) is appropriate. The periodic verification approach needs only to focus on evaluating which valve design features and fluid conditions are covered by the data, which are covered by extension and which are not covered. Figure 19 shows a flow chart of the classification process. The coverage of compressible flow, elevated temperatures, high flow rates and flashing flow is discussed below under unbalanced disk globe valves.

Periodic Verification of Unbalanced Disk Globe Valves

Figure 20 shows a typical unbalanced disk globe valve. The stem moves a disk toward or away from a seat to close or open the valve. The DP acts across the disk. Resistance to disk motion comes from packing friction, the effect of pressure pushing the stem out of the valve (stem rejection), and the effect of DP acting across the disk area. Only the last term is affected by the presence of flow and DP across the valve during its stroke, but it is not susceptible to degradation. Accordingly, testing in the JOG MOV PV Program was performed to confirm the absence of degradation.

From the test data, the DP thrust was determined and expressed as a valve factor, for those strokes where the DP thrust opposed disk motion (closing strokes for valves with underseat flow and opening strokes for valves with overseat flow). In all cases, the valve factor was observed to be stable. Figure 21 shows the results for eight globe valves in water flow < 120ºF. (In Figures 21 and 22, values are not shown on the y-axis because they are not needed to understand the observed trends.) The average valve factor across three tests is observed to be stable. Although there are minor test-to-test changes for specific valves, these changes are within the measurement uncertainty. Figure 22 shows the results for three valves in steam flow. Two valves, marked UG07 and UG13, show stable results. (In the case of UG07, there are two curves because the valve factor was calculated at two points in the stroke.) One valve, UG14, shows an increase in the closing direction from the first to the third test. The measurement uncertainty is large for these tests because the valve DP was very small when the valve seated. This result occurred because the downstream piping depressurized slowly as the valve closed and was still nearly at full pressure when the valve seated. To address this shortcoming in the test, the valve factor was determined with an alternate method using the opening data (self-actuating stroke), which had the full DP. The result, as shown on Figure 22, is a stable valve factor.

Because no degradation was observed in unbalanced disk globe valves, a periodic verification approach of periodic static testing (Class A or B) is appropriate. The periodic verification approach needs only to focus on evaluating which valve design features and fluid conditions are covered by the data, which are covered by extension and which are not covered. Figure 23 shows a flow chart of the classification process. The unbalanced disk globe valve tests covered incompressible water flow and steam flow; steam results are consistent with water flow. No results were obtained for flashing flow. The maximum flow velocity in the balanced and unbalanced disk globe valve tests (86 ft/sec, based on the seat area) was used to set an applicability limit on the method.

Summary

1. The JOG MOV PV Program is being used by the vast majority of US nuclear power plants to implement MOV periodic verification and to determine the potential degradation in required thrust or torque for gate, globe and butterfly valves.
2. A key component of the JOG PV Program is in-plant valve testing. The testing is now complete and there are repeat test data from 176 valves.
3. For all four valve types tested, there is no age-related degradation (i.e., no increases in required thrust or torque due only to the passage of time without DP stroking).
4. Gate valves are susceptible to service-related degradation only when they have low initial valves factors, either due to disassembly of the valve or due to little or no DP stroking in service. For these valves, valve factor increases tend to occur progressively up to a plateau level as the valve accumulates DP strokes. Valves that are set up using a justified valve factor do not need to consider

increases. Valves that are set up using a valve factor susceptible to increase need to add a margin allowance to cover future increases in required thrust.

5. Butterfly valves have no service-related bearing friction degradation. Bronze bearings have stable friction in treated water and in untreated water when the valve has a bearing hub seal. Bronze or 300 series stainless steel bearings in untreated water without a hub seal show significant friction variations, with no trend. Non-metallic bearings show small friction variations in both treated and untreated water. Valves that are set up using a justified bearing friction coefficient do not need to consider the effect of variations. Valves that are set up using a friction coefficient susceptible to variations need to be justified by DP testing or set up to cover the variations.

6. For balanced disk globe valves and unbalanced disk globe valves, there is no service-related degradation in required thrust. For balanced disk globe valves, the DP thrust component is small and the valve factor is stable. For unbalanced disk globe valves, testing confirmed a stable thrust in both water and steam. In balanced disk globe valves, service in untreated water can lead to thrust variations, not related to DP thrust, that come and go. It appears that these variations are due to particulates interfering with disk motion.

7. A periodic verification approach has been defined and justified, based on the results of the JOG MOV PV Program. The approach classifies valves according to their susceptibility to increases in required thrust or torque. Valves that are set up in a manner that is not susceptible to degradation have periodic static testing at a frequency depending on risk and margin. Valves that are susceptible to increases either have specified margin allowances to be added or need to have periodic DP testing.

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Table 1. Periodic Verification Intervals for the JOG MOV PV Program

Risk Ranking(2)	PV Test Interval (years) for…		
	Low Margin(1)	Medium Margin(1)	High Margin(1)
High Risk	2	4	6
Medium Risk	4	8	10
Low Risk	6	10	10

Notes:

1. Criteria for MOV Margin Categories

    Low Margin: JOG MOV PV Margin < 5%

    Medium Margin: 5% ≤ JOG MOV PV Margin < 10%

    High Margin: 10% ≤ JOG MOV PV Margin

2. Criteria for Risk Categories

    High Risk
    Medium Risk
    Low Risk

    } Based on Owners' Group or utility-specific criteria.

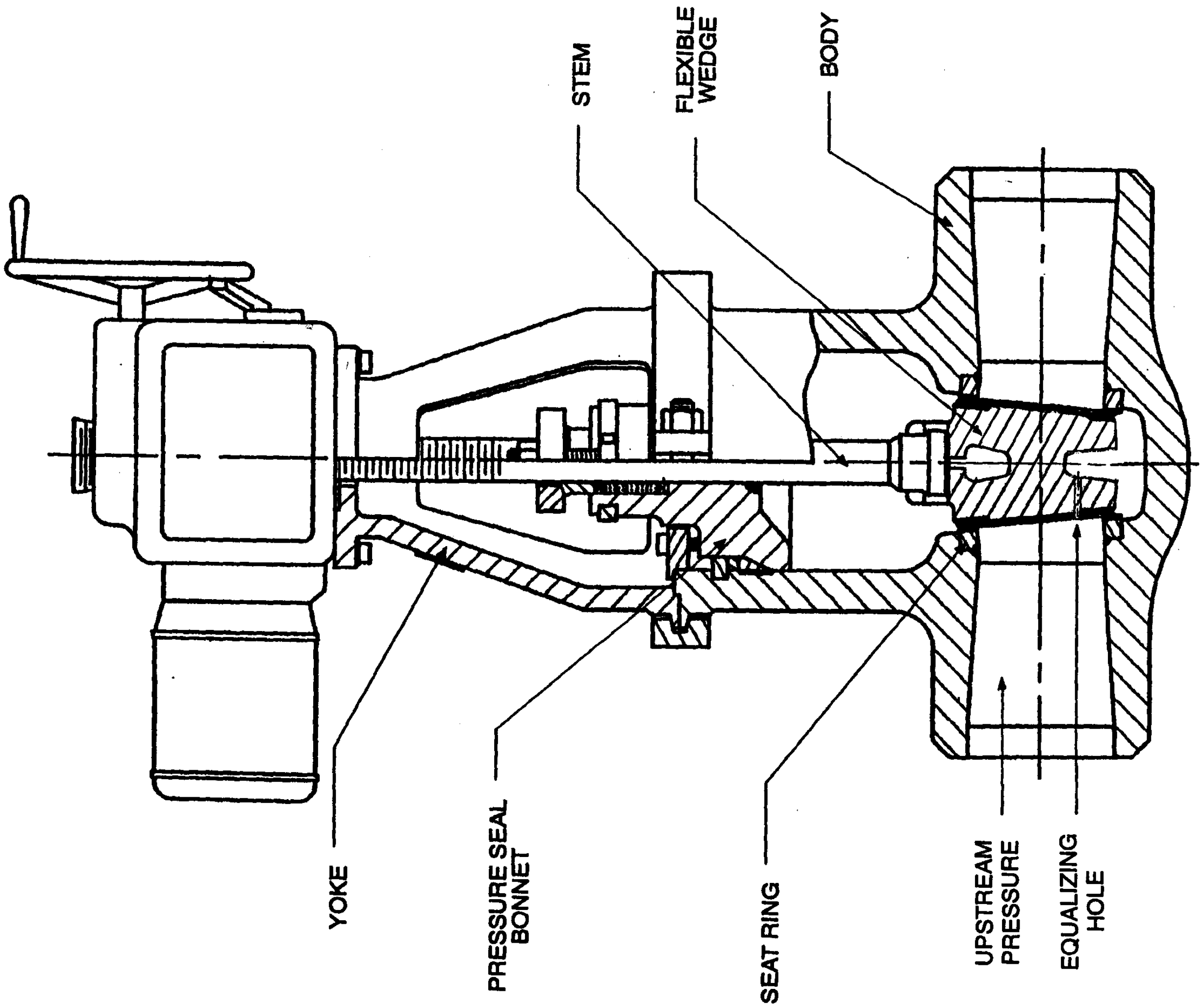
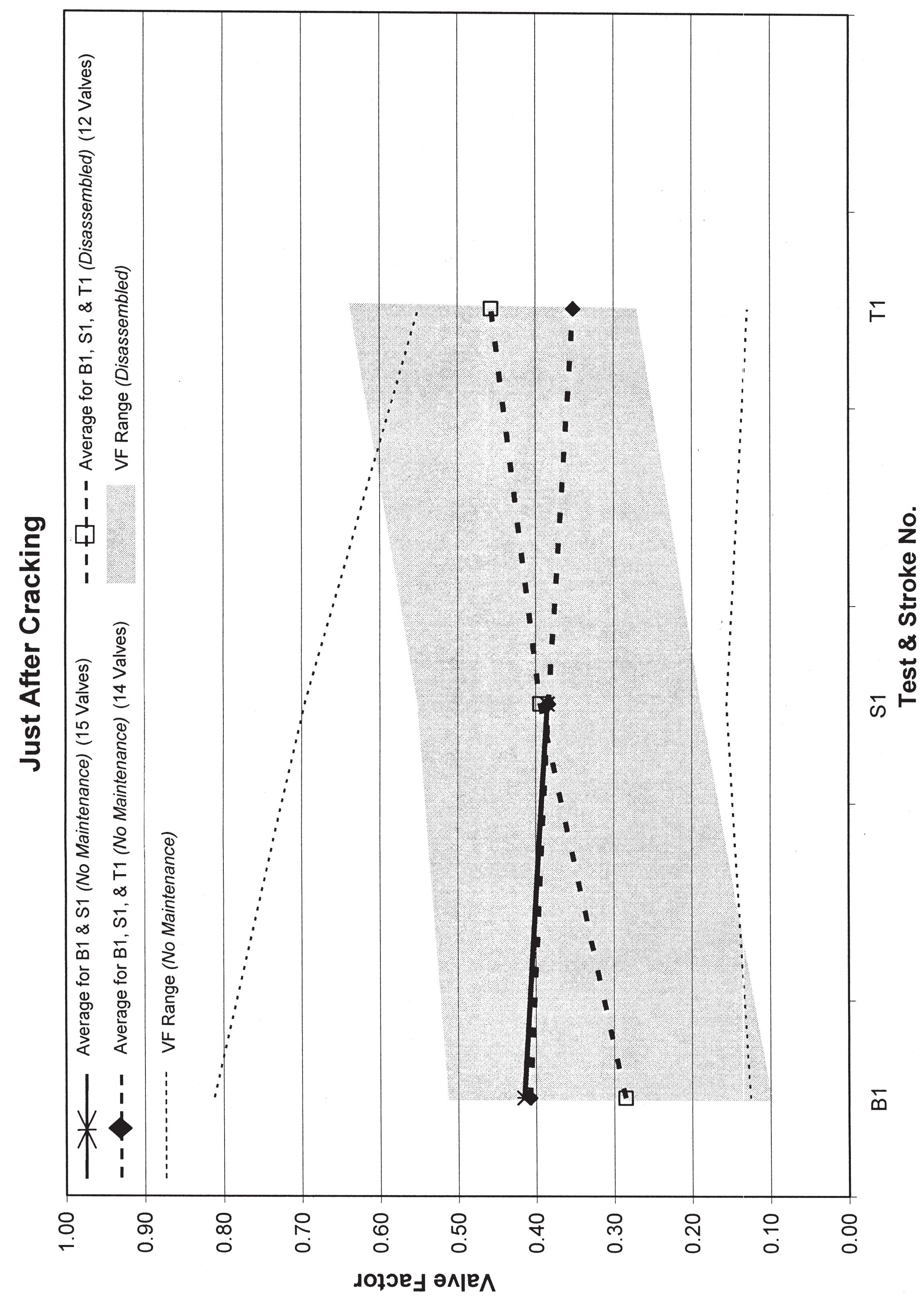


Figure 1. Typical Gate Valve

Figure 2. Opening Valve Factors (Just after Cracking) for Gate Valves with Stellite Seats in Treated Water Systems with 1 to 4 DP Strokes Between Tests



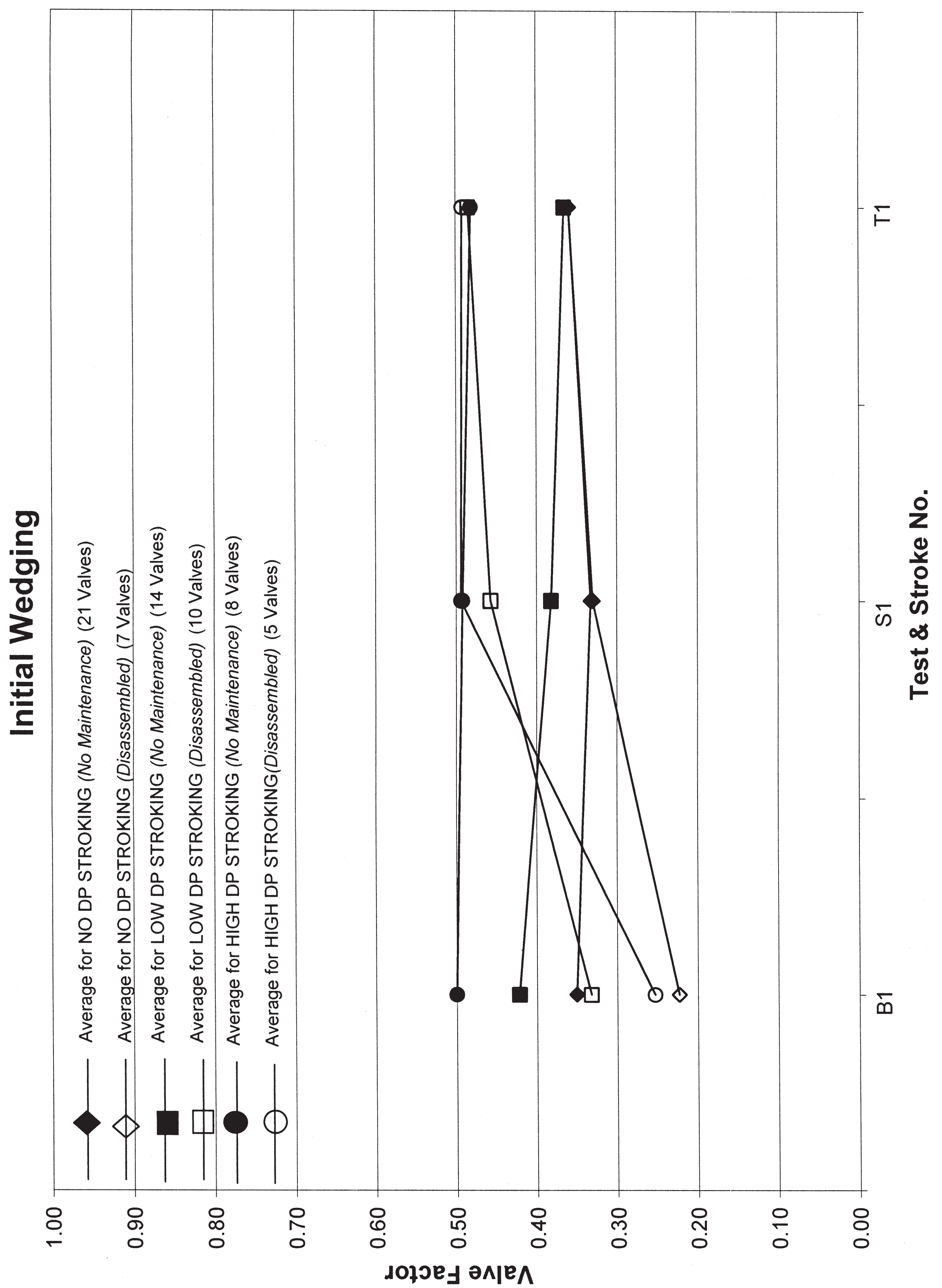
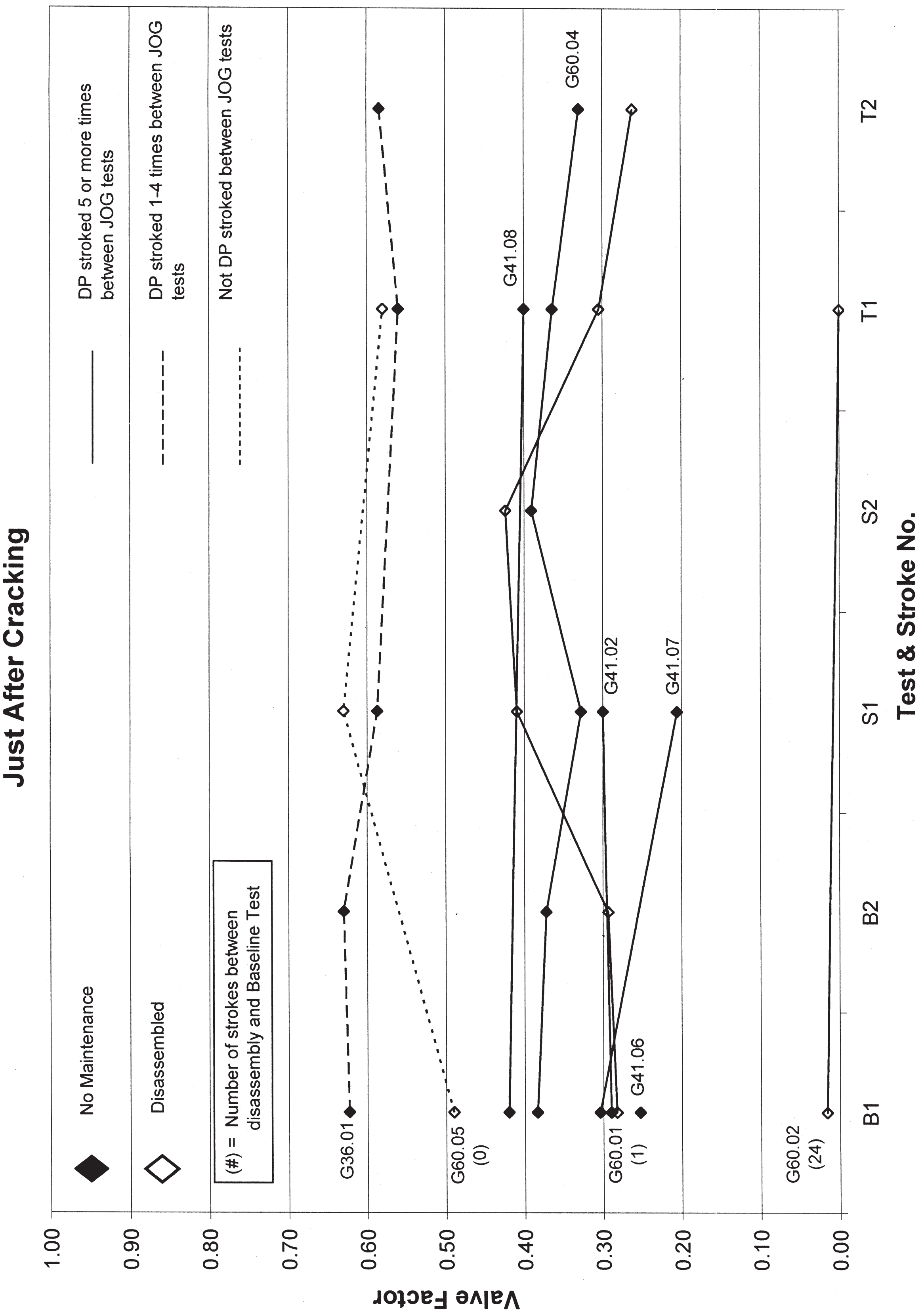


Figure 3. Valve Factors for Disassembled and Non-disassembled Valves with Stellite Seats and Different Amounts of DP Stroking

Figure 4. Opening Valve Factors (Just after Cracking) for Gate Valves with Stellite Disks and Seats in Steam



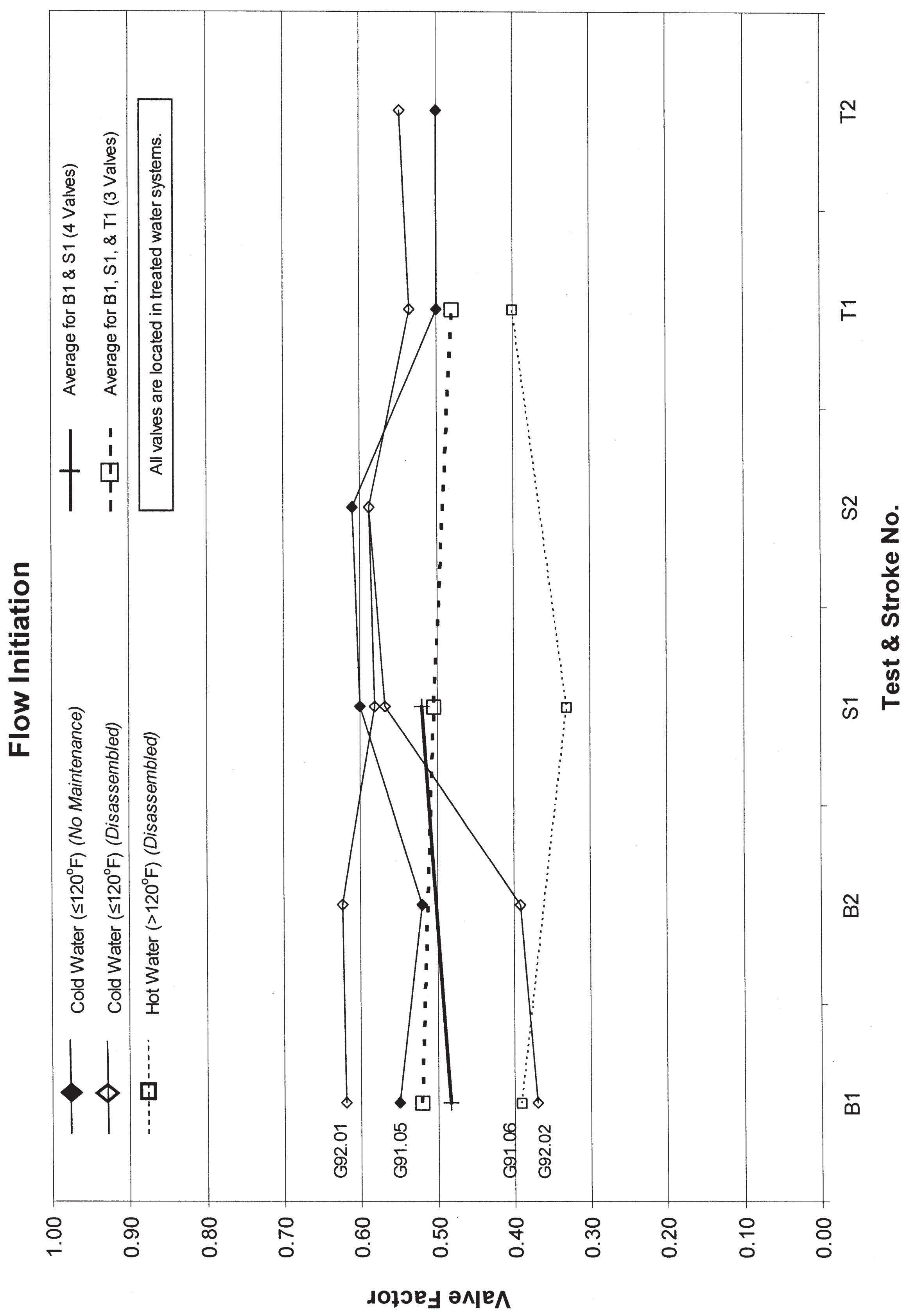
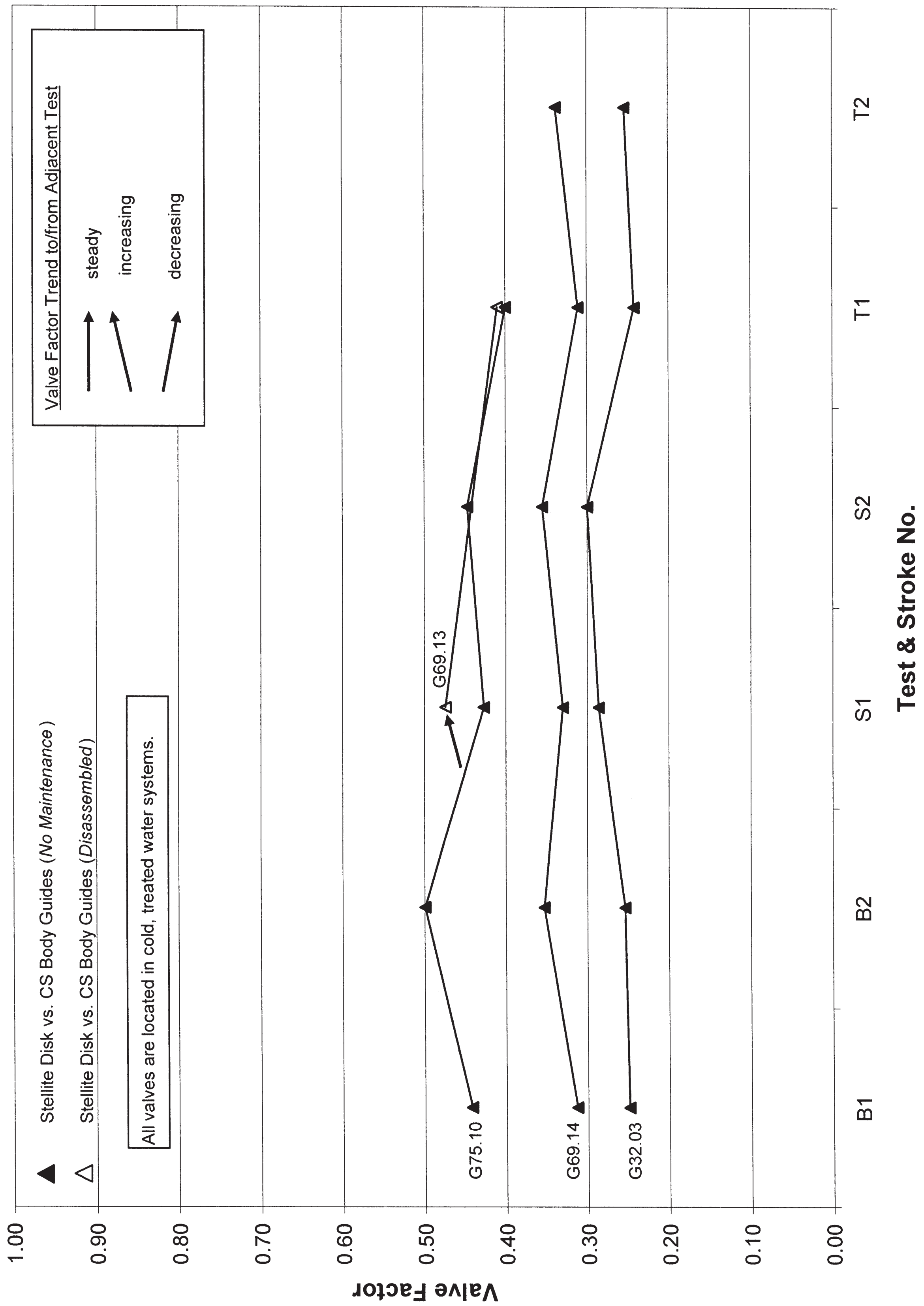


Figure 5. Opening Valve Factors (Flow Initiation) for Gate Valves with 400 Series Stainless Steel Disk and Stellite Seat Ring Faces

Figure 6. Guide Valve Factors for Gate Valves with Stellite Disk Guides and Carbon Steel Body Guides



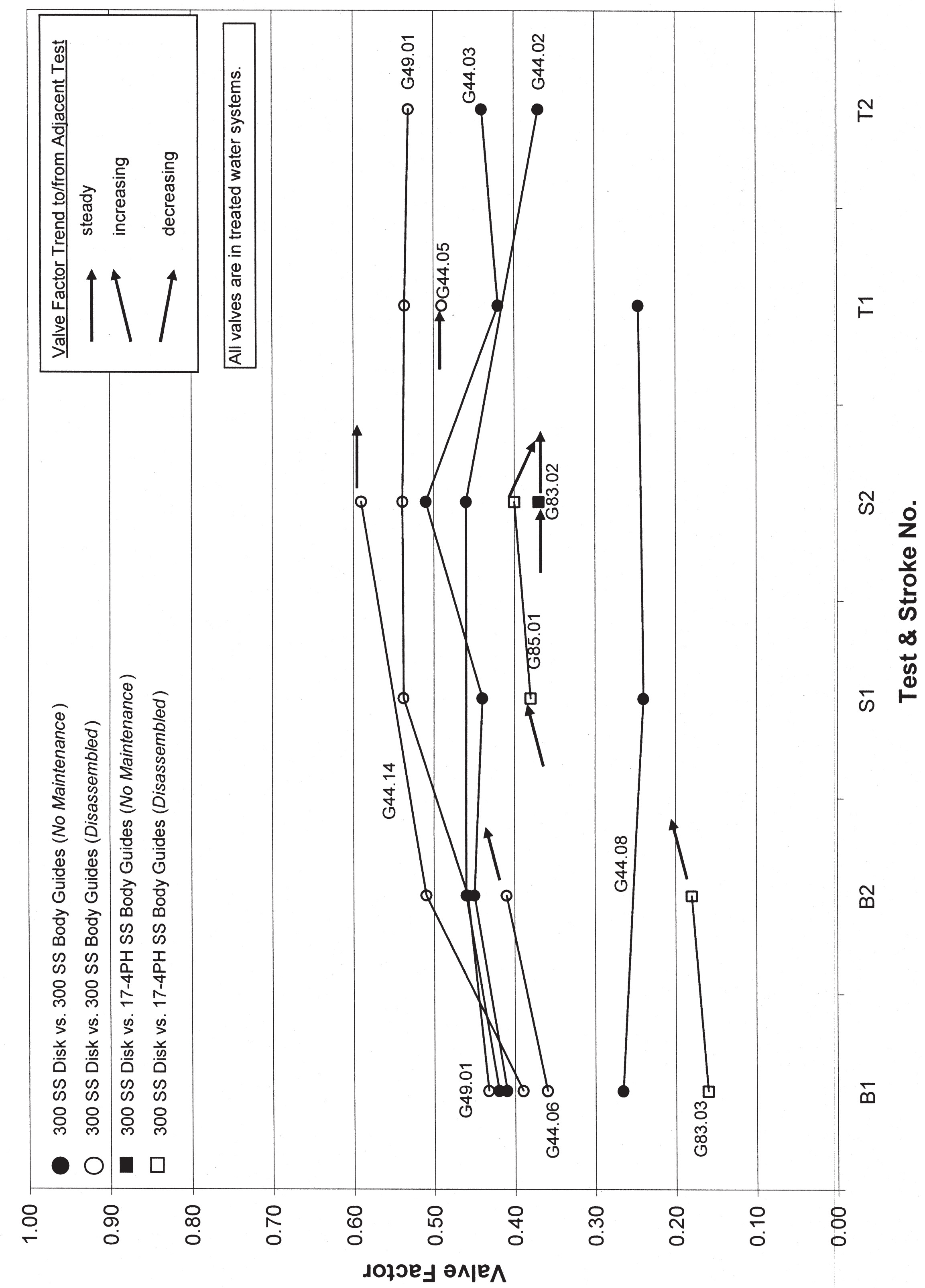
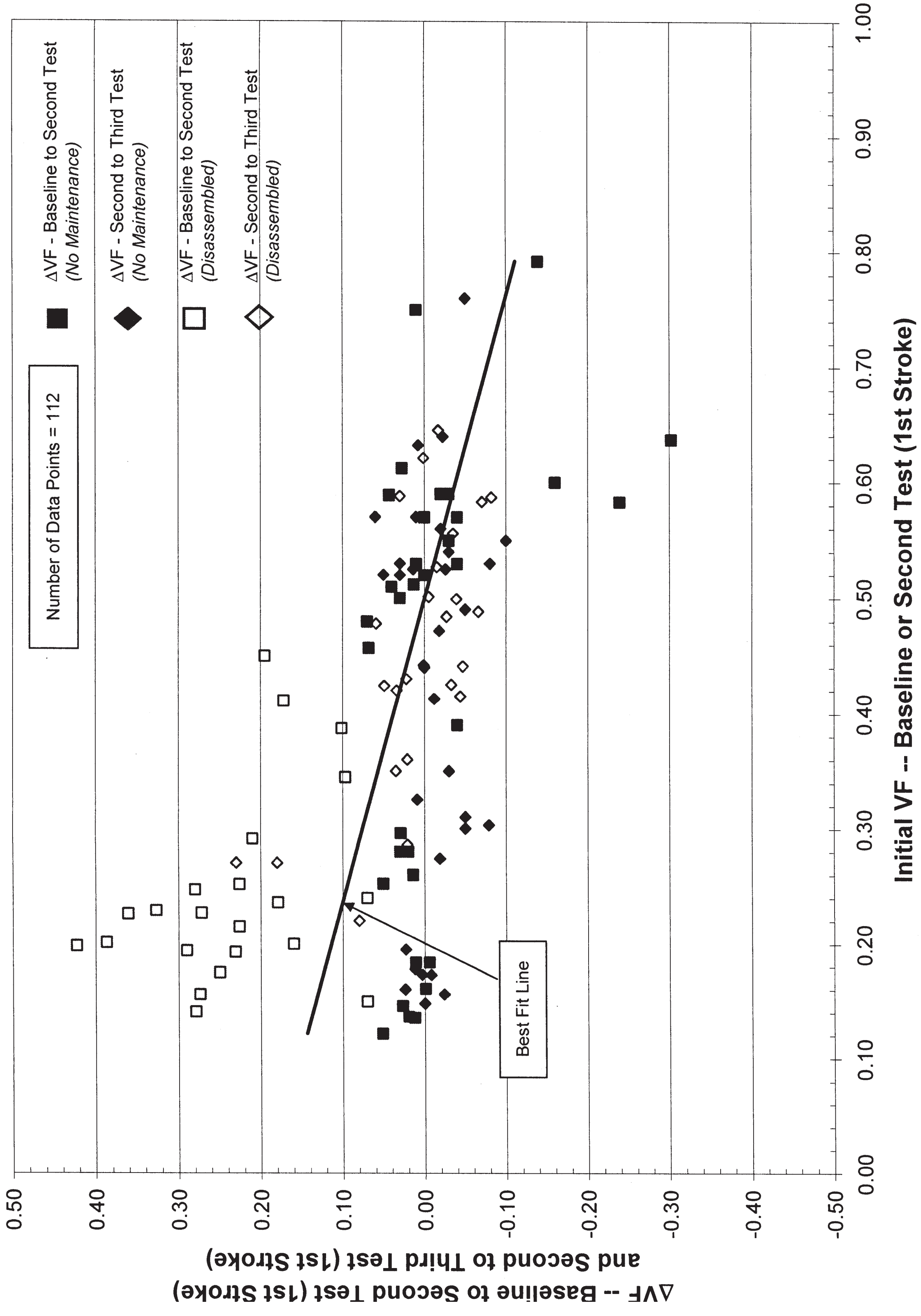
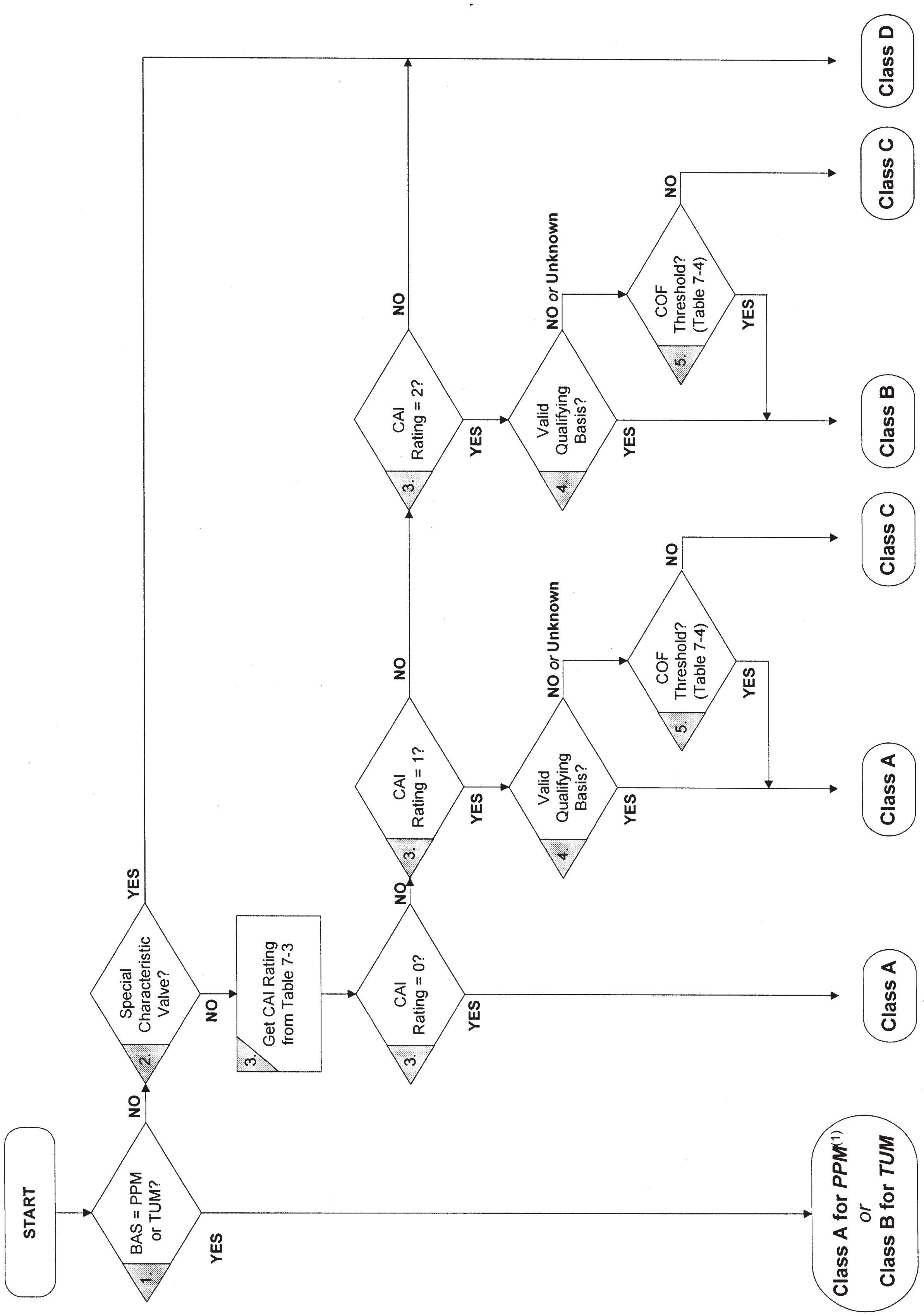


Figure 7. Guide Valve Factors for Gate Valves with 300 Series Stainless Steel Disk Guide Faces and Either 300 Series or 17-4 PH Stainless Steel Body Guide Faces

Figure 8. Change in Disk-to-Seat Friction Coefficient vs. Initial Friction Coefficient for Valves with Self-mated Stellite Seats in Water





Note 1: PPM evaluations beyond the nominal applicability limits (i.e., "best available data") are considered Class B. See Gate Valve Method Step 1 for additional discussion.

Figure 9. Gate Valve Classification Flow Chart

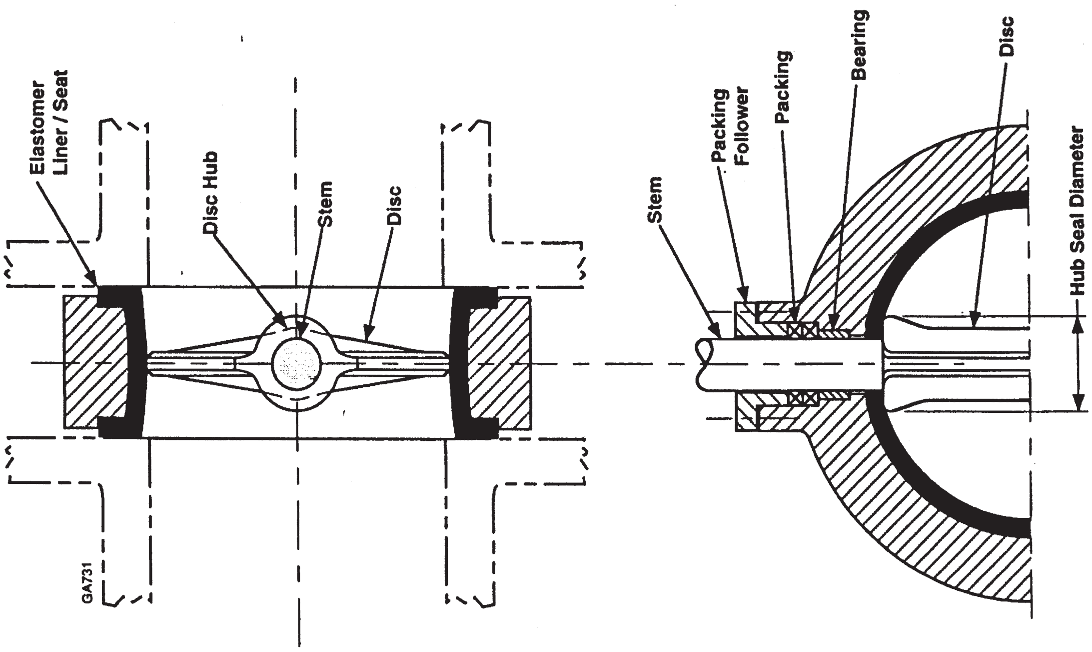


Figure 10. Typical Butterfly Valve (Symmetric Shaft Design)

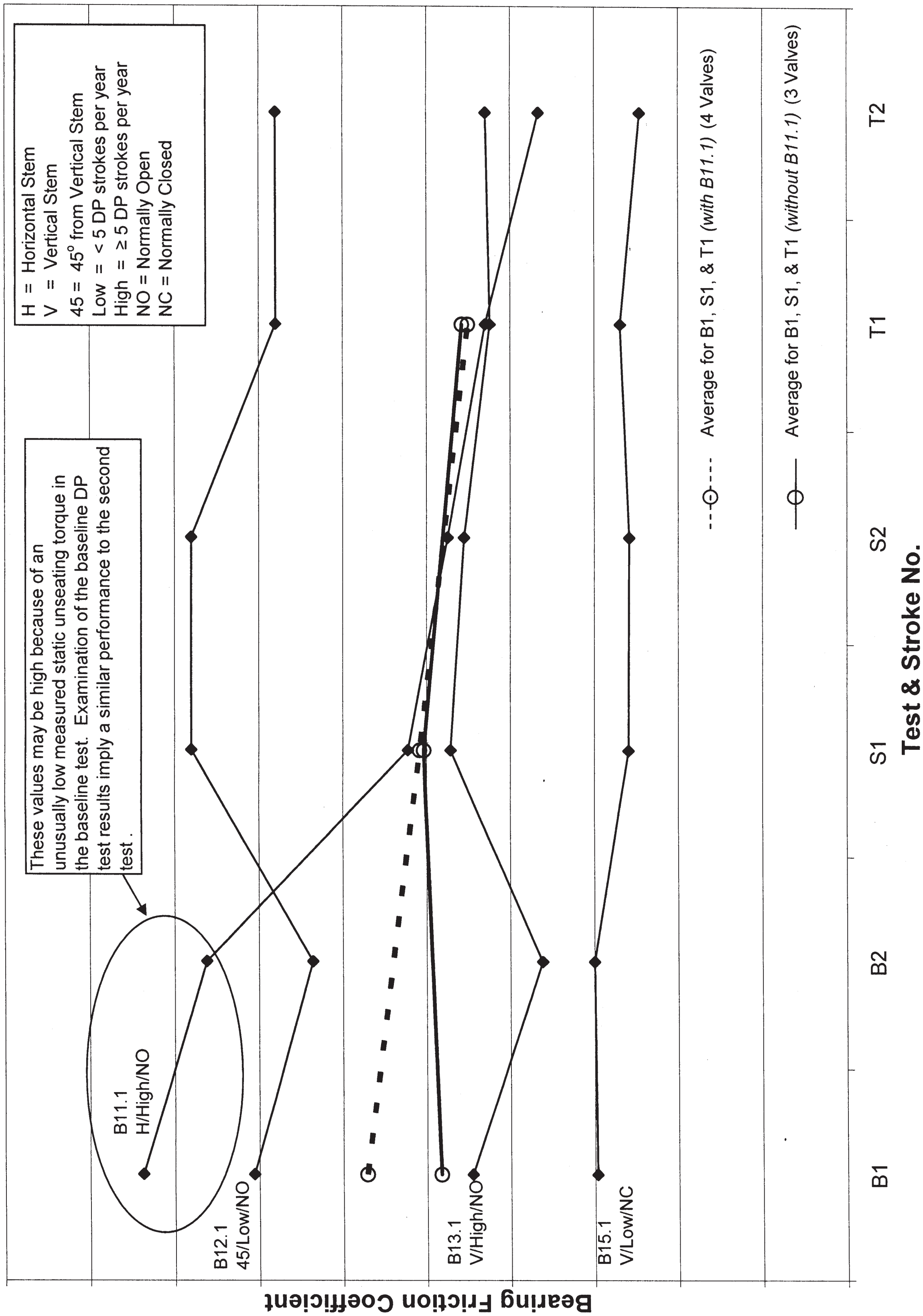
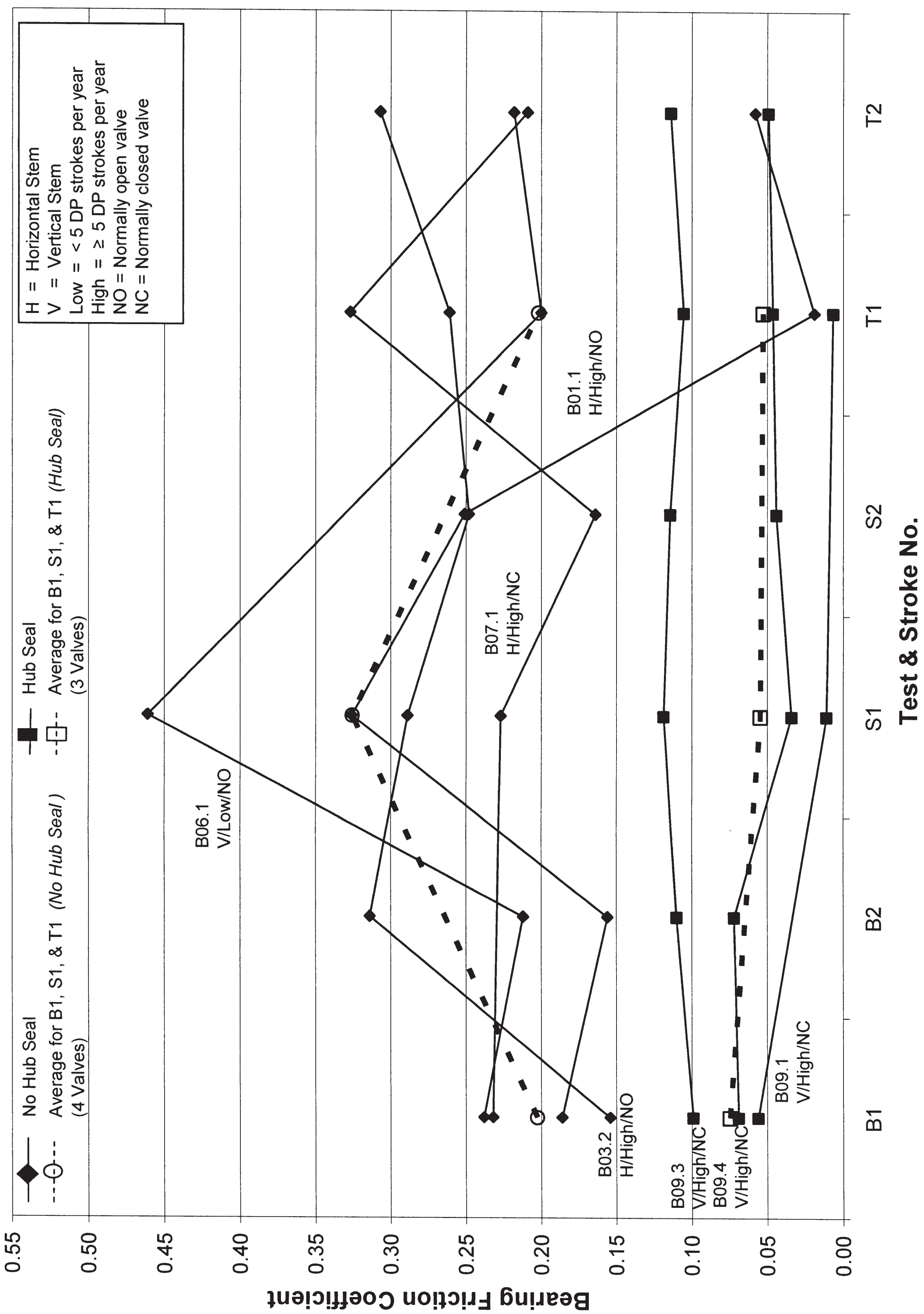


Figure 11. Bearing Friction Coefficient for Bronze Bearings in Treated Water Systems

Figure 12. Bearing Friction Coefficient for Bronze Bearings in Untreated Water Systems



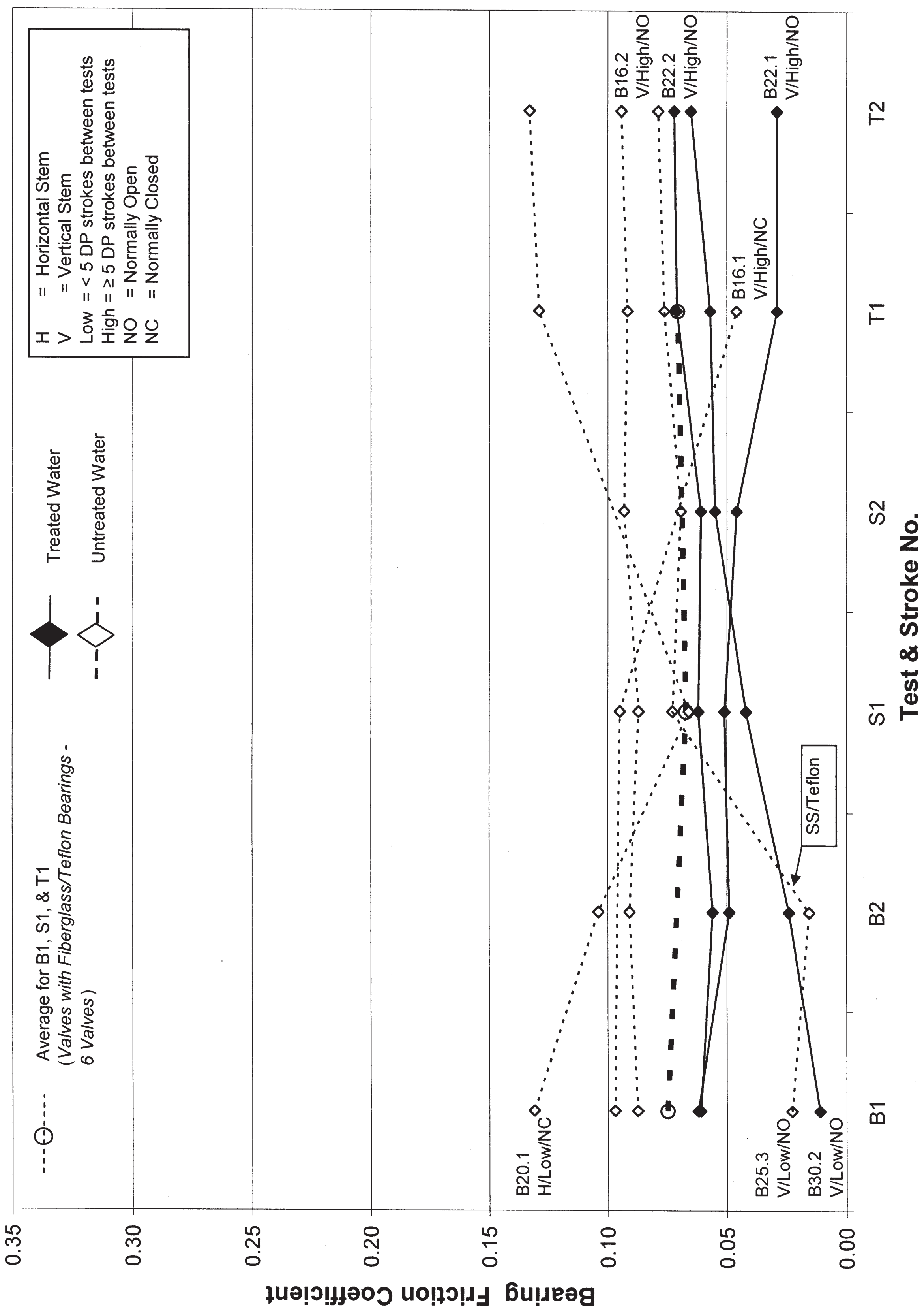
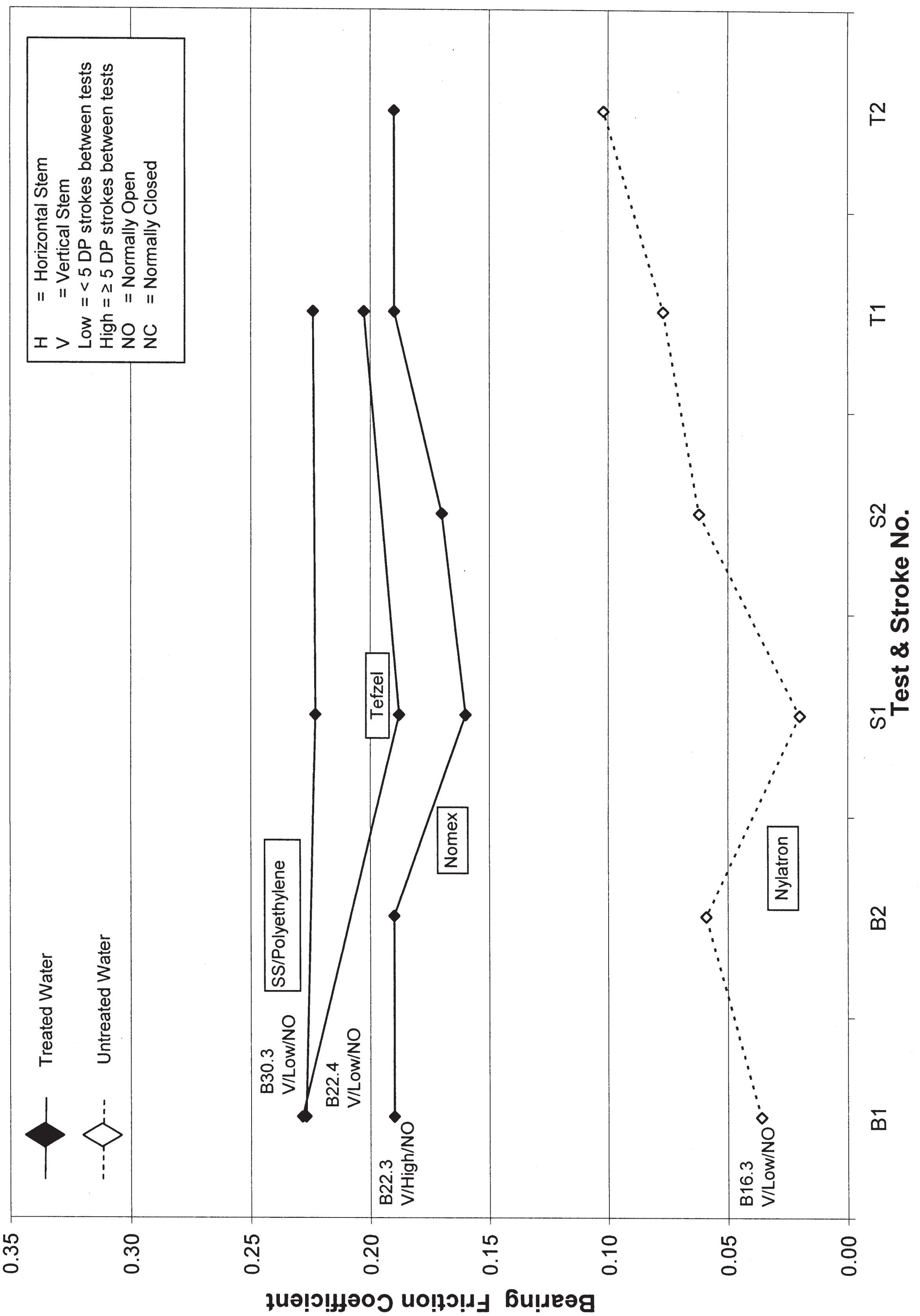
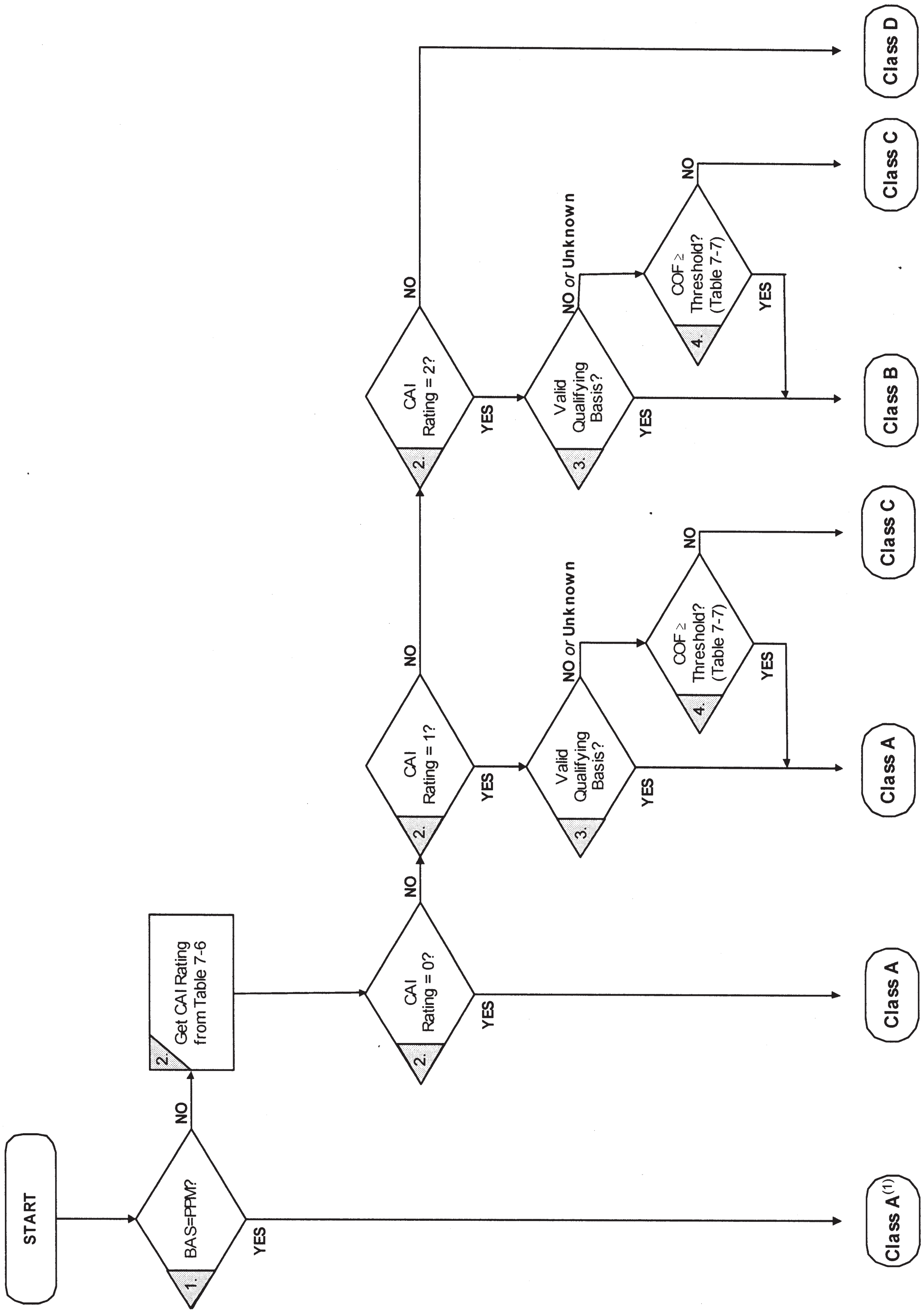


Figure 13. Bearing Friction Coefficient for Teflon-lined Bearings in Water Systems

Figure 14. Bearing Friction Coefficient for Non-metallic Bearings (other than Teflon) in Water Systems

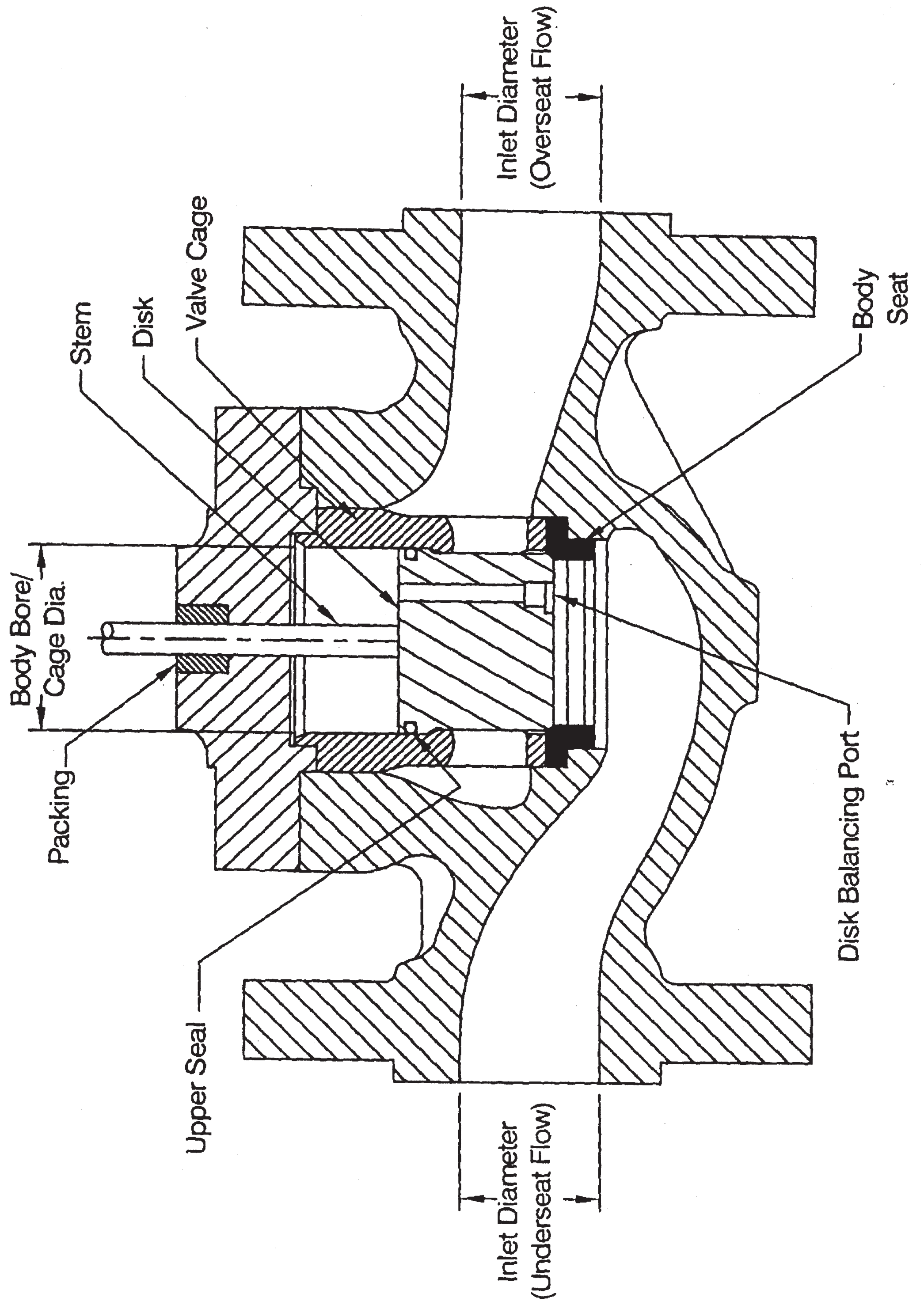




Note 1: PPM evaluations beyond the nominal applicability limits (i.e., "best available data") are considered Class B. See Butterfly Valve Method Step 1 for additional discussion.

Figure 15. Butterfly Valve Classification Flow Chart

Figure 16. Typical Balanced Disk Globe Valve



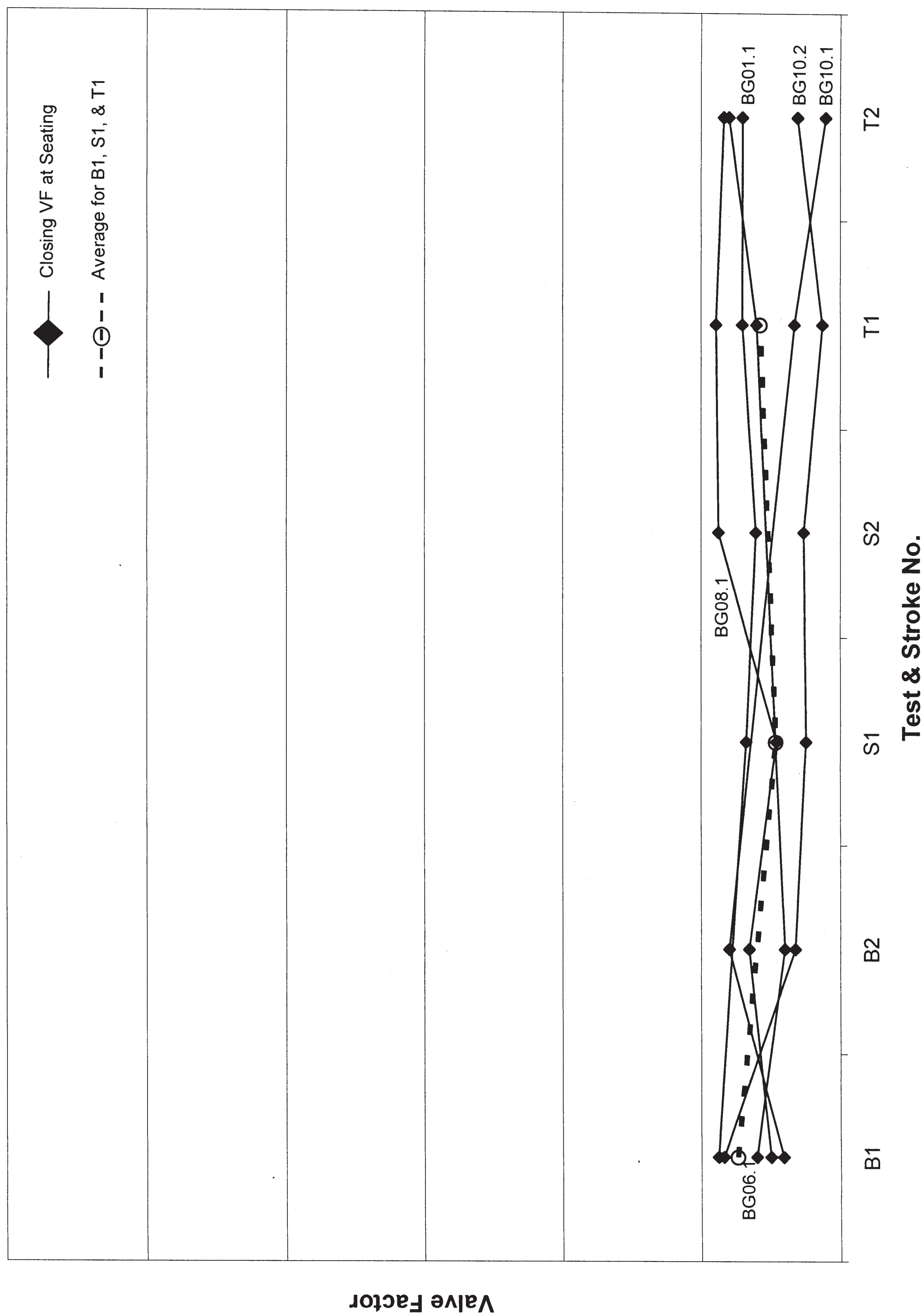
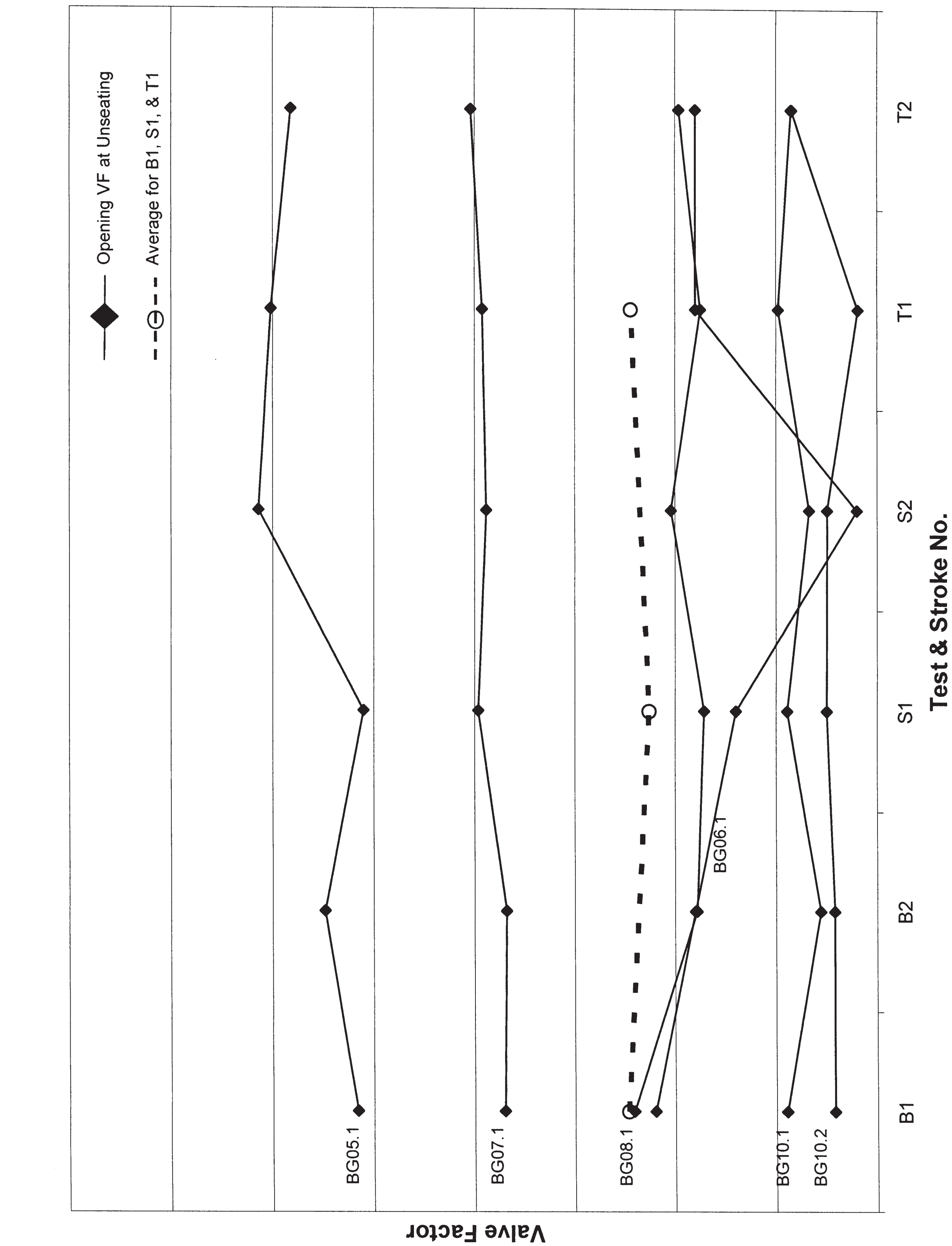
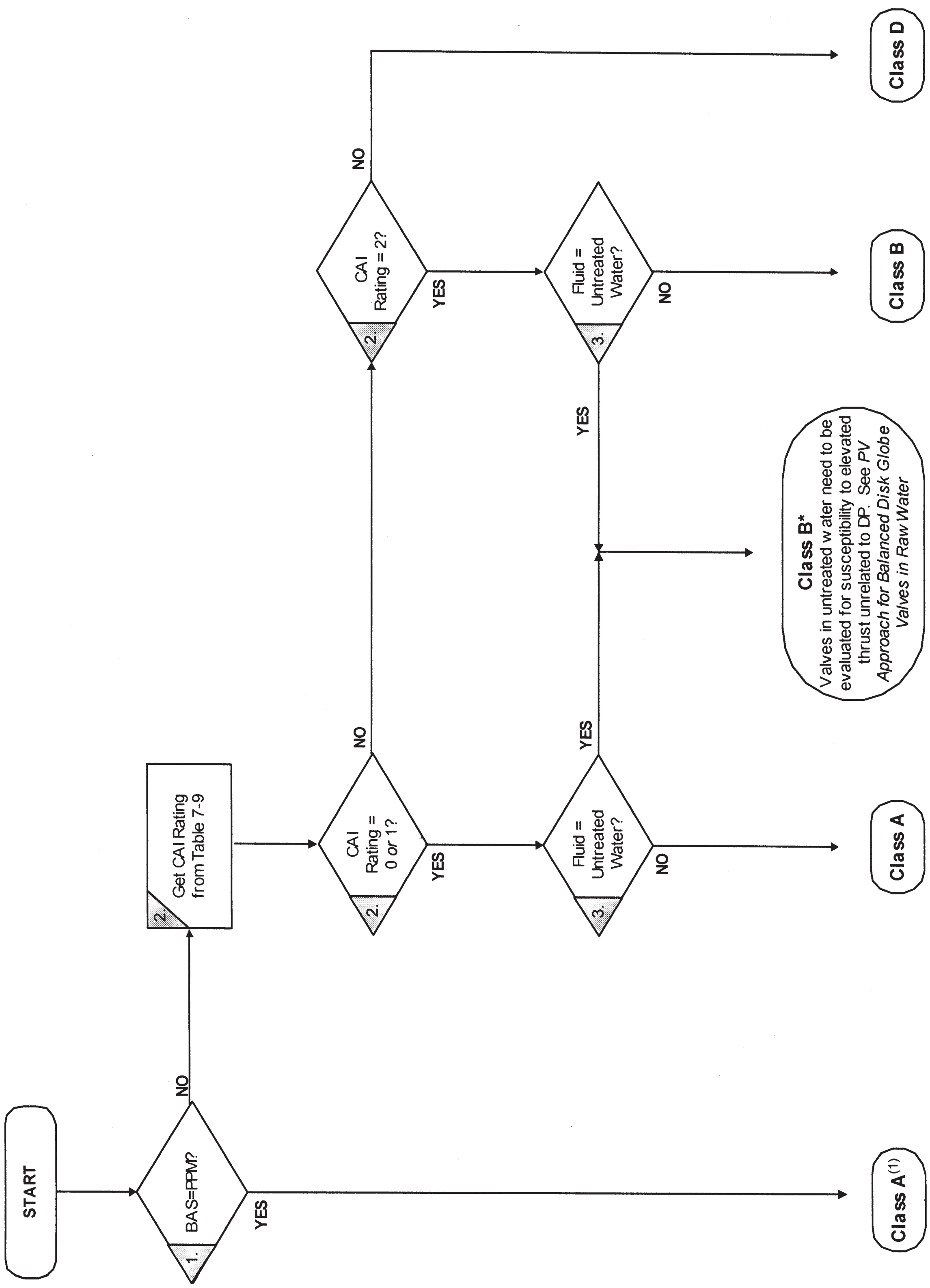


Figure 17. Valve Factors for Closing Strokes of Balanced Disk Globe Valves

Figure 18. Valve Factors for Opening Strokes of Balanced Disk Globe Valves





Note 1: PPM evaluations beyond the nominal applicability limits (i.e., "best available data") are considered Class B. See Balanced Globe Valve Method Step 1 for additional discussion.

Figure 19. Balanced Disk Globe Valve Classification Flow Chart

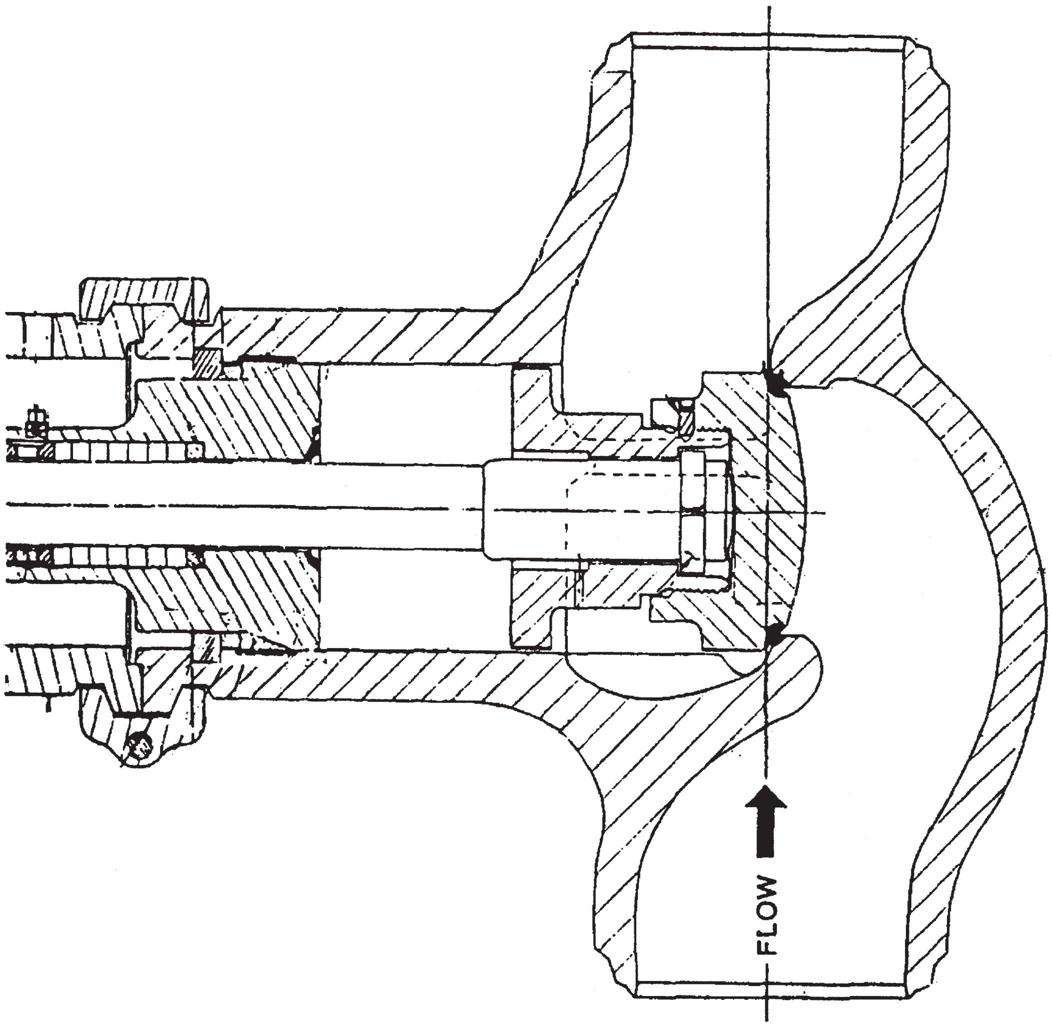


Figure 20. Typical Unbalanced Disk Globe Valve (Underseat Flow)

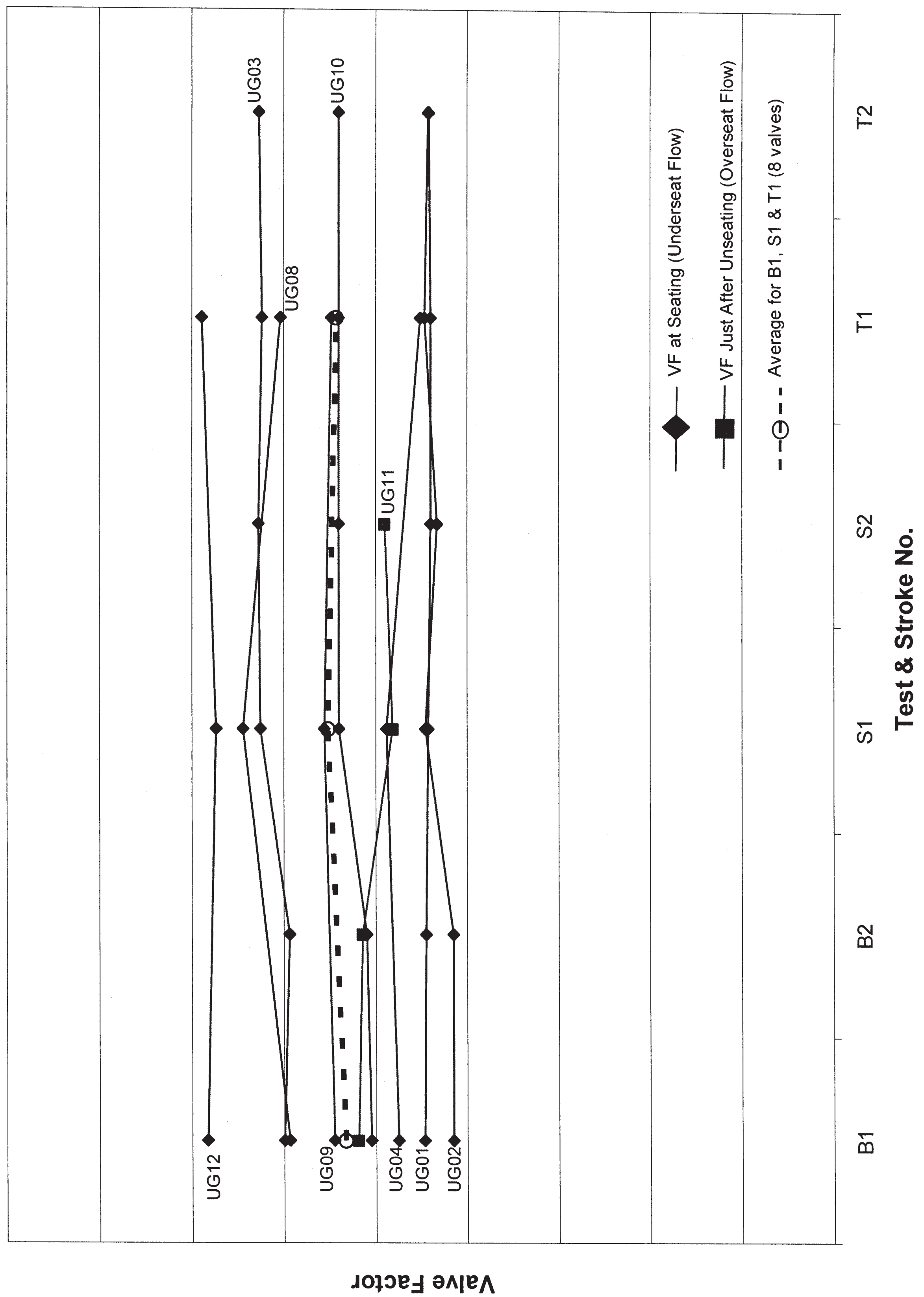
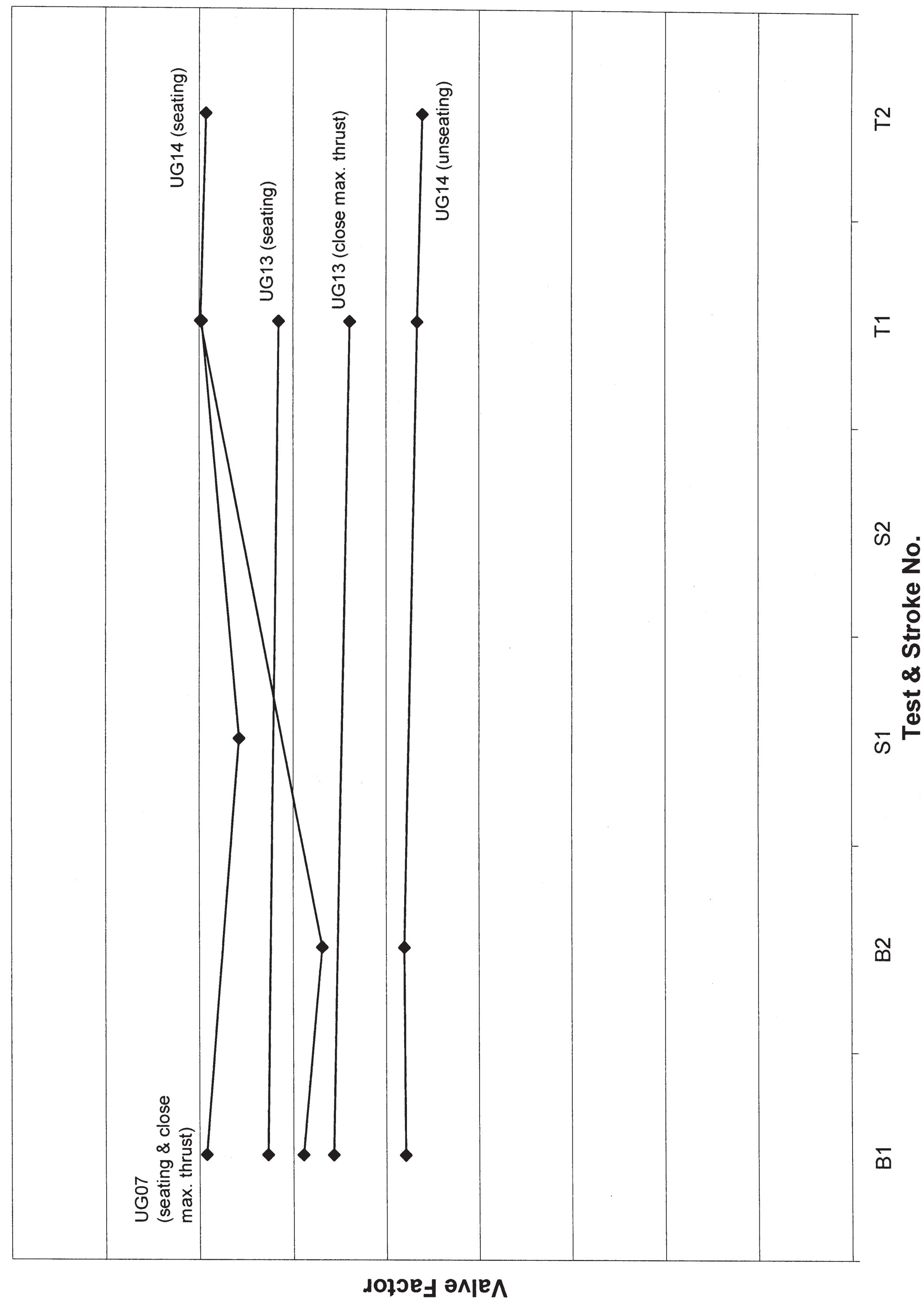
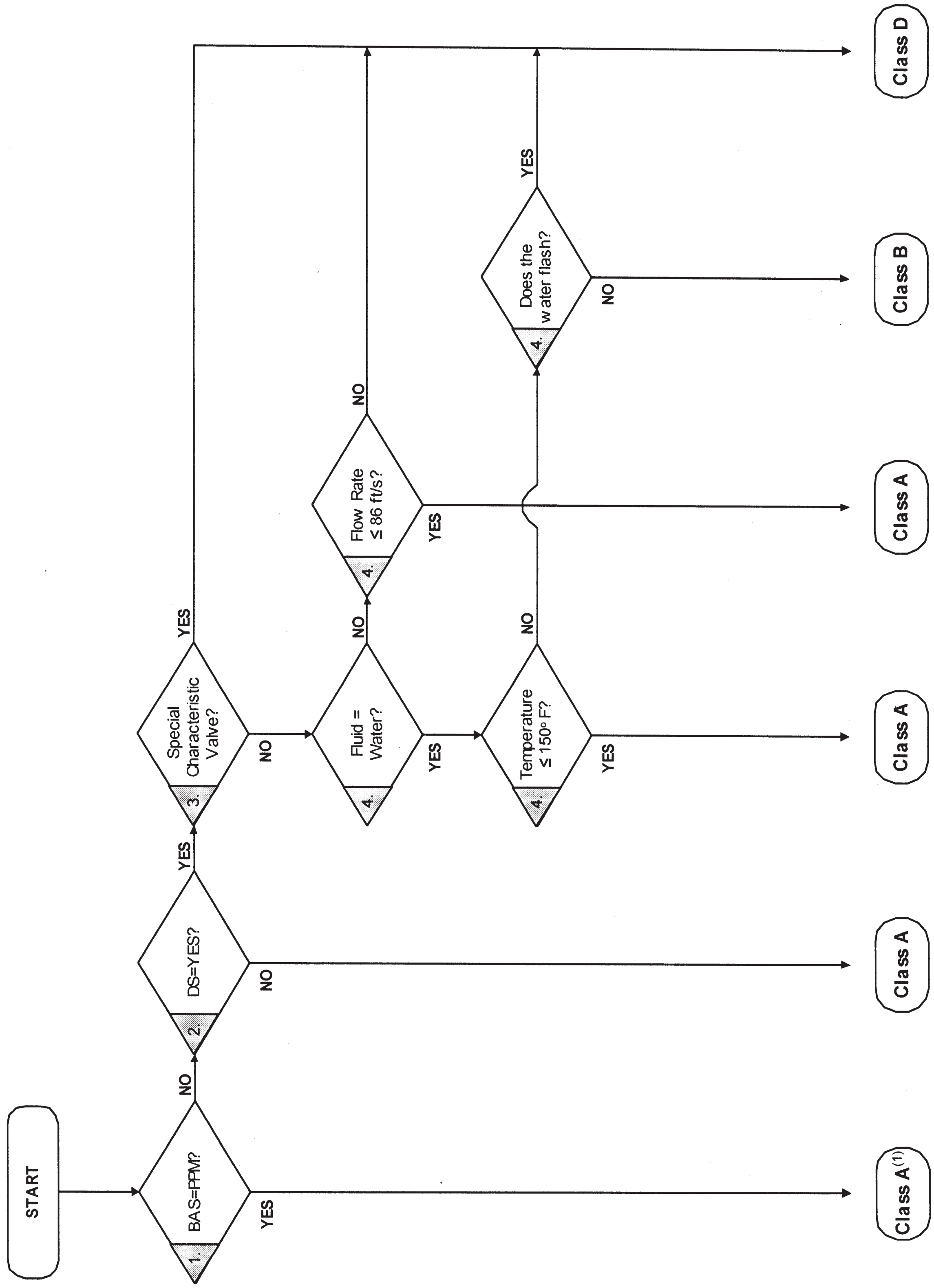


Figure 21. Valve Factors for Unbalanced Disk Globe Valves in Water Flow

Figure 22. Valve Factors for Unbalanced Disk Globe Valves in Steam Flow





Note 1: PPM evaluations beyond the nominal applicability limits (i.e., "best available data") are considered Class B. See Unbalanced Globe Valve Method Step 1 for additional discussion.

Figure 23. Unbalanced Disk Globe Valve Classification Flow Chart

EPRI MOV Stem Lubricant Test Program

Frictional Performance of Exxon Nebula and MOV Long Life in a Stem Lubrication Application

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ABSTRACT

This paper reports initial results of a program to assess the frictional performance of various lubricants in a motor-operated valve (MOV) stem lubrication application. The program will assess the effects of stem loading time-history and temperature on stem friction for a total of ten stem lubricants. Results for the first two lubricants tested (Exxon Nebula and MOV Long Life) are presented herein.

INTRODUCTION

Motor-Actuator Operation

Figure 1 shows the internal components in a typical motor-operated valve actuator. When the motor is activated, a motor pinion gear turns a splined shaft that turns a worm, rotating a worm gear that is keyed to a stem nut resulting in rotation of the nut. The actuator stem is driven up or down by the ACME threaded connection to the stem nut. The torque imparted to the stem by the stem nut is reacted below either by a torque reaction arm built into the valve or by the disk within the valve against the valve seats. As more torque is produced (due to resistance of linear motion occurring in the valve) the worm is driven to the right compressing the spring pack (a series of Belleville washers). When a pre-selected displacement of the spring pack is reached, the torque switch is tripped deactivating the motor. The stem/stem-nut connection converts rotational motion to linear motion or torque to thrust. The friction coefficient at the stem/stem-nut interface is a critical factor in determining the efficiency with which torque is converted to thrust and therefore the thrust that can be produced for a given torque switch setting.

Ambient Temperature Effects

Over the past 14 years, the Electric Power Research Institute (EPRI) and the industry have conducted testing to determine the MOV actuator stem/stem-nut coefficient of friction (COF) and changes in stem friction with loading condition (rate-of-loading) for several stem lubricants and stem/stem-nut configurations. All safety-related MOVs are currently setup based on stem friction coefficients measured in these tests. These data were generally obtained at room temperature conditions. Recent testing sponsored by the U.S. Nuclear Regulatory Commission (NRC) Office of Nuclear Regulatory Research and conducted by the Idaho National Engineering and Environmental Laboratory (INEEL) (References 1 and 2) has shown that for some lubricants, dynamic stem friction coefficients can increase with temperature (20-30% increase in friction with a temperature increase from 21 to 121 degrees C (70 to 250 F). Such an increase in stem friction coefficient would result in a proportionate reduction in the thrust output of MOV actuators (under dynamic loading) at their current control (torque) switch settings.

A review of the INEEL test program completed by EPRI concludes that the testing was conducted using sound testing methods and that the results are accurate for the conditions tested. However, the review also concludes that direct application of the results to industry valves may be difficult for a variety of reasons. Examples include: repeatable performance was not always established prior to varying test parameters, the stem remained in compression at all times unlike many valves that unload (redistributing the grease at the stem/stem-nut interface) during opening strokes, and all tests were conducted under simulated DP loading conditions with no intervening static strokes that would also tend to redistribute the grease. The EPRI review recommends a more comprehensive test program to assess potential temperature effects on stem to stem-nut friction that addresses the issues discussed above.

Stem Loading Effects

In addition, Exxon Nebula grease that is used extensively as a stem-to-stem nut lubricant is no longer being produced. As the current stem friction and rate-of-loading specifications for many plants with this lubricant are based on extensive plant unique tests, moving to a new lubricant may require a reassessment of stem friction and rate-of-loading effects for such plants. A new lubricant (MOV Long Life) has been approved for use as a gearbox grease replacement for Nebula and appears to be an excellent candidate for a replacement for Nebula as a stem lubricant. Data are needed to assist utilities in justifying the switch from Nebula to MOV Long

Life as a stem lubricant without additional plant unique testing to reestablish their stem friction and rate-of-loading specifications.

Rate-of-Loading is defined as the percentage reduction in actuator output thrust at torque switch trip (TST) on a closure stroke, between a static (no differential pressure on valve disk) and a dynamic (flow and differential pressure on valve disk) condition. Research conducted in the mid 1990s determined that the rate-of-loading phenomenon is caused by a squeeze film effect at the stem/stem-nut thread interface. During a dynamic closure stroke, the loading on the valve and resulting thread contact stress increases gradually, and the grease at the stem/stem-nut interface is slowly squeezed out of the threads resulting in most of the stroke occurring with metal-to-metal contact or in a boundary lubrication condition. The resulting friction coefficient is generally in the 0.1 to 0.15 range. In contrast, during a static closure stroke, the threads are relatively lightly loaded for all but the last 100 milliseconds (ms) of the stroke when the valve disk reaches the seat. At this point the load increases very quickly to the point when the torque switch trips. In this very short seating period, the grease has insufficient time to fully squeeze out of the thread interface resulting in a momentary hydrodynamic lubrication condition. This can result in friction coefficients in the 0.03-0.07 range. This reduction in friction coefficient in the static test results in more thrust being produced at torque switch trip (TST) during a static closure stroke than in a dynamic stroke. In addition, during a dynamic stroke, the friction coefficient just prior to seating can be somewhat higher than at torque switch trip. This additional effect is accounted for by the addition of margin in torque switch set-up values.

Utilities utilize diagnostic equipment to measure the thrust output of the actuator at TST. The torque switch is set to obtain the required thrust at TST during a static test (when the stem friction coefficient can be reduced due to rate-of-loading). Many utilities have conducted extensive static and dynamic tests on the same valves to develop a statistical specification that conservatively defines the plant rate-of-loading effect for their valve population. This effect must be accounted for when defining the required thrust at TST.

The magnitude of the rate-of-loading effect can be affected by several factors including stem and stem nut fit up, surface roughness, and geometry and type of lubricant. Current rate-of-loading specifications account for all factors listed above except switching to a new lubricant.

Accordingly, data are needed to establish the effect of temperature on the dynamic (boundary lubrication) stem friction coefficient for stem lubricants currently in use (including MOV Long Life). In addition, data are needed to assess potential differences in room temperature rate-of-loading effects between Exxon Nebula and MOV Long Life.

TEST SYSTEM DESCRIPTION

An actuator test fixture has been designed (see Figures 2 and 3) to allow time-dependent loading of the stem during operation simulating both static and dynamic conditions at a variety of stem/stem-nut grease temperatures. The test fixture is located at EPRI's Charlotte facility. Many components of the test fixture are the same as those used in the rate-of-loading research program conducted on behalf of EPRI by Battelle Columbus in the early 1990s. The test stand includes a new surplus Limitorque actuator (SMB-0, 25 horsepower (HP), 230/460 volts-alternating current (VAC) motor) with MOV LongLife Grade 1 grease in the gearbox and Mobil grease 28 in the limit switch compartment. The actuator gear ratio is chosen to provide a stem speed ranging from 31.75 to 63.5 centimeters per minute (cm/min) (12.5 to 25 inches per minute) depending on the lead of the stem tested. The test stand allows application of a time dependent load history simulating both dynamic and static strokes in both the opening and closing directions, i.e., the stem will go from compression to tension as stroke direction is reversed.

The actuator stem is driven up or down by the rotation of the stem nut within the actuator. The lower end of the stem is threaded and keyed into an adaptor hub. The adapter hub is bolted to an anti-rotation device that has two arms with roller bearings at each end. The stem torque is reacted by machined faced bar stock beams attached to a simulated valve yoke assembly.

Four stop beams are bolted to the bottom of the anti-rotation device. During actuator closure strokes, the lower two beams contact stops bolted to the base plate. Contact with the base plate stops simulates gate or globe valve hard seat contact. After contact with the base plate stops, the thrust load increases rapidly until the torque switch trips deactivating the actuator.

Passive Hydraulic System

The purpose of the hydraulic cylinder is to provide resistance to motion of the actuator stem simulating loading that may occur during valve operation under either static (no flow or differential pressure) or dynamic (flow and differential pressure) conditions. In the original rate-of-loading test program conducted by Battelle, hydraulic pressure to drive the cylinder was provided by a hydraulic pump and associated control system. In the new design, no hydraulic pump will be required. Resistance to motor actuator stem motion will be produced by controlling the flow of fluid from one side of the piston to the other using a rectifier block and a proportional relief valve.

The passive hydraulic system is employed to simulate valve operation. The entire system is pressurized to 1.38 MegaPascals (200 pounds per square inch gage (psig)) to ensure that hydraulic fluid does not cavitate in low-pressure portions of the circuit. Figure 4 shows operation of the hydraulic system simulating valve-closing operation. As the actuator moves the stem, the hydraulic fluid is pushed from the left side of the cylinder into the rectifier block. The check valves within the block direct the fluid upward and out of the block at the top where it passes through a filter and into a proportional relief valve. The relief valve flow is controlled by a signal from the data acquisition computer. The relief valve limits the flow; thereby, building pressure on the left side of the cylinder to resist motion of the actuator. The system can provide constant low loads (simulating packing load) as low as 4448.2 Newtons (1000 lbs) and time-varying loading up to 146,790 Newtons (33,000 lbs). A cylinder by-pass loop with a manual valve is included to allow development of very low packing loads as required. The flow exits the relief valve at a low pressure and enters a water-cooled heat exchanger, and then enters the right side of the cylinder. Experience in use of the system indicates that minimal heating of the hydraulic fluid occurs obviating the need for active cooling.

The system includes high and low pressure side gages, a hydraulic fluid thermometer, and an accumulator to ensure that the system operates at a constant backpressure regardless of fluid temperature increases and/or fluid seepage.

Applying a voltage from 0 to 10 volts DC to the valve's control amplifier can vary the relief pressure of the proportional relief valve. The amplifier then converts the control signal to a pulse width modulated current that drives the solenoid to the desired position. The signal to control the relief valve position is programmed by the operator using the Labview program developed to support the test program.

The system has a pressure capability of 15,569 MegaPascals (3500 psi). In operation, the system pressure does not exceed 8896.4 MegaPascals (2000 psi).

Stem Heating System

A 20.32 cm (8 inch) long cartridge heater is inserted into a hole drilled down each stem centerline and is used to heat the area of the stem nut and grease for the elevated temperature tests. The heater is controlled in closed loop using a type K thermocouple spot welded to each stem just below the bottom of the stem nut when the stem is in the up (retracted) position. The thermocouple provides feedback to a solid-state temperature controller that brings the stem to the programmed temperature without overshoot. Differences in temperature between the thermocouple location and the middle of the stem nut (highest temperature region) are accounted for in setting the target stem temperature. A separate effects test was conducted to establish such temperature differences at each of the temperature levels to be tested. The stem temperature was stabilized to the target temperature to within +/- 2.8 degrees C (5 degrees F) for 15 minutes.

INSTRUMENTATION AND DATA ACQUISITION

The actuator and test system are instrumented to allow measurement of actuator output thrust and torque, cylinder stem position (same as actuator stem position), stem temperature in the area of the stem nut, torque switch activation, and spring pack displacement. All measurements will be recorded using a high-speed data acquisition system except for stem temperature. Stem temperature measurements will be made and recorded manually. Table 1 lists the instrumentation and data acquisition rates for each measurement.

Thrust and Torque

Thrust and torque are measured using a Crane Torque Thrust Cell (TTC). Two Vishay 2311 Signal Conditioning Amplifiers are used to provide excitation voltage and amplify torque and thrust signals. Once amplified, the thrust and torque signals are routed to a BNC Connection box and then cabled to a National Instruments 6036E Multifunction DAQ Card. This card interfaces with the PC and Labview Software. Labview software is used to acquire and analyze the data as well as send the control voltage to the proportional relief valve.

Torque Switch Trip

A key measurement is the time of torque switch trip. This is the reference point for comparing the rate-of-loading characteristics of the stem/stem-nut. Torque switch trip is not the point at which the actuator stops putting out torque and thrust. It is the point (time) at which the current to the switch is lost (indicating that the selected spring pack displacement has been reached and the torque switch has opened) and the relay it holds closed begins to open. Once that relay has opened, additional time passes before the contactors "drop out" de-energizing the motor. Even then, the actuator continues to generate output torque and thrust due the inertia of the motor and gearing within the actuator until the disk finally comes to a stop against the seats (or, in this case, against the stops). This results in a measurable increase in output thrust and torque after the torque switch has opened. Such increases in the thrust/torque need to be considered in evaluating the structural capability of the actuator, valves and, in our case, test system. However, it is not relevant to the rate-of-loading phenomenon that relates only to the thrust and torque output at the moment of torque switch trip. Accordingly, a method is needed to precisely determine the moment when the torque switch actually opens.

A custom torque switch trip circuit was designed by Battelle in the original test program and is being implemented in this program as well. The circuit generates a TTL signal (Transistor-Transistor Logic step change in voltage) at the initiation of the opening of the torque switch contacts. The circuit generates and latches (holds) the signal when the frequency of the electric motor-starter holding coil current changes from 60 hertz (Hz). The input to the circuit is from a current probe hooked around a loop of 10 coils of wire connected to the torque switch close terminal.

TEST MATRIX

Data are recorded only during closure strokes. In addition, data are recorded on static closure strokes only under room temperature conditions. The opening strokes are conducted only for the purpose of repositioning the stem to the open position and redistributing the grease at the stem/stem-nut interface. Opening strokes do not involve torque switch trip (the actuator is limit controlled in the opening direction) and, therefore, provide no meaningful quantitative information with regard to the rate-of-loading (ROL) phenomenon. Further, data need not be collected for elevated temperature static closure tests as all in-plant diagnostic testing used to set torque switches is conducted at room temperature.

Each stem-lubricant combination undergoes a test sequence involving 99 total strokes. Data are recorded for 30 closure strokes, and 25 dynamic and 5 static strokes. Each test sequence includes confirmation of stability in the thrust at torque switch trip followed by a set of 5 static and 5 dynamic closure strokes conducted at room temperature to assess rate-of-loading effects. These tests are followed by 5 dynamic closure strokes at nominal temperatures of 130, 190, 250 and 70 degrees F. Low load static strokes are conducted between dynamic strokes to reposition the stem and redistribute the lubricant. Each lubricant is tested on three stems (A, G and I) as detailed in Table 2.

RESULTS

Rate-of-loading

Figure 5 compares the observed rate-of-loading performance of each stem for each lubricant tested. Each column shown in Figure 5 represents the average rate-of-loading for the 5 sets of static and dynamic tests conducted on each stem-lubricant combination. All data shown are for room temperature conditions.

The rate-of-loading percentages shown are computed using the following equation:

ROL % = (Thrust at TST Static –Thrust at TST Dynamic) X 100 / Thrust at TST Dynamic

Stem A and Stem I exhibited significant ROL, while Stem I showed minimal ROL.

With the exception of the data labeled Nebula *, no significant differences in rate-of-loading performance were observed between MOV Long Life and Nebula. The first test series conducted on Stem A using Nebula resulted in the data represented by the column labeled Nebula *. As these data were not consistent with the data obtained from the other two stems, this series was repeated. The data from the repeat series was consistent with the performance observed on the other stems.

Effect of Stem Temperature

Each lubricant (Nebula and MOV Long Life) was tested on three stems (A, G and I) at four nominal temperature levels (70, 130, 190 and 250 degrees F). Five dynamic tests were performed at each temperature level with intervening static strokes conducted between dynamic strokes. The stem coefficient of friction was calculated for each stroke using the corrected thrust and torque and appropriate stem dimensional information in the following equation:

$$\text{Stem COF} = (0.96815 * d * (24 * 3.14 * SF - L)) / (24 * SF * L + 3.14 * d^2)$$

Where:

d = Pitch Diameter = Stem O. D. – ½ * Pitch (inches)

SF = Absolute value of the Stem Factor = Corrected Torque (Ft-lbs)/Corrected Thrust (lbs)

L = Stem Thread Lead (inches)

The grease on the stem in the area of the stem nut was heated using a cartridge heater inserted into a hole drilled down the stem centerline to a point coincident with the stem nut location when the stem is in the up (retracted) position. All heating is conducted with the stem in this retracted position.

The test system was capable of heating Stems A and G to 121 C (250 F) but was only able to reach a peak stem temperature of 113 C (235 F) for Stem I. This still allowed adequate definition of the effect of grease temperature on stem coefficient of friction.

Figure 6 shows the effect of stem temperature on dynamic friction for Nebula for each of the three stems tested. Each data point represents the average of the 5 COF values obtained in the 5 tests conducted at each temperature. Each COF value is the maximum recorded during the last second prior to hard seat contact during dynamic closure strokes. The stem thread pressure during this portion of the stroke is approximately 110 MegaPascals (16,000) psi. Stem thread pressure is calculated assuming that the entire thrust is being applied to a single thread.

As shown in Figure 6, minimal change (of the order of 5 %) in stem COF is evident for Stems I and G. Stem A shows a more significant increase (of the order of 20 %) in COF from 21 to 121 degrees C (70 to 250 degrees F).

Figure 7 shows the effect of temperature on stem coefficient of friction for MOV Long Life on each of the three stems tested. Increasing the stem temperature from 21 to 121 degrees C (70 to 250 degrees F) resulted in increases in stem COF ranging from 13 to 26 % depending on the stem tested.

Figures 8 through 10 compare temperature effects for Nebula and MOV Long Life exhibited on stems A, G and I, respectively. The most significant temperature effects were for Stem A and Stem I. Stem G consistently exhibited lower temperature effects for both lubricants. The effect of temperature on stem friction is slightly greater for MOV Long Life compared to that for Nebula for the stems tested.

The stem coefficient of friction returned close to, and in many cases lower than, its original room temperature value after the stem was cooled back to room temperature.

On two tests, the torque switch tripped prior to the stem reaching the hard stop. These were tests on Stem I, MOV Long Life at temperatures of 88 and 113 degrees C (190 and 235 degrees F), respectively. Stem I exhibited consistently high COFs for both lubricants tested.

CONCLUSIONS

The objectives of this phase of the project are to:

1. Compare the rate-of-loading performance of Nebula EP-1 and MOV Long Life, and
2. Assess the effect of temperature on the dynamic coefficient of friction at the stem/stem-nut interface for Nebula and MOV Long Life.

With regard to the first objective, these tests show no significant difference in rate-of-loading performance between Nebula and MOV Long Life.

With regard to the second objective, the results for these tests indicate some increase in stem friction coefficient for both Nebula and MOV Long life with MOV Long life exhibiting a somewhat greater effect than Nebula. Previous testing by INEEL (References 1 and 2) on different stems showed minimal effects of temperature on stem friction for these lubricants. It is concluded that temperature effects on stem friction can occur for these lubricants and that the magnitude of such effects is stem dependent.

REFERENCES

1. K.G. DeWall, et al, “Performance of MOV Stem Lubricants at Elevated Temperature,” NUREG/CR-6750, October 2001.

2. K.G. DeWall, et al, “MOV Stem Lubricant Aging Research,” NUREG/CR 6806, March 2003.

Table 1

Test System Instrumentation

Measurement	Transducer Selected	Full Scale Calibrated Range	Transducer Accuracy	Data Acquisition Rate
Stem Torque	Crane TTC RC	+/- 1170 ft-lbs	+/- (2% of Reading + 0.5% Full Scale)	1000 samples/sec
Stem Thrust	Crane TTC RC	+/- 40,000 lbs	+/- (1% of Reading + 0.5% FS)	1000 samples/sec
Stem Temperature	Fluke Model 52 Thermometer	-328 to +2501 Deg F	+/- 0.05% of Reading + 0.5 Deg F	N/A-Manual recording
Stem Position	MTS Temposonics APM	0-6 inches	+/- 0.05% FS	1000 samples/sec
Torque Switch Current	Fluke Clamp-on Probe	N/A - Used for timing only.	N/A	1000 samples/sec
Limit Switch Current	Fluke Clamp-on Probe	N/A - Used for timing only.	N/A	1000 samples/sec
Torque Switch activation	Fluke Current Sensor/ TST Circuit	N/A - Used for timing only.	N/A	1000 samples/sec

Table 2

Stems and Stem-Nuts Tested

Stem	Stem Geometry (inches)	Stem Material	Stem Nut Threaded Length (inches)	Stem Velocity (inches/min)	Rate of load increase after hard seat contact (lbs/sec)
A	2 x ¼ x ½	17-4 Ph	3.88	25.0	185,000
G	2 x ¼ x ½	410 SS	3.25	25.0	185,000
I	1.75 x ¼ x ¼	17-4 PH	6.00	12.5	108,800

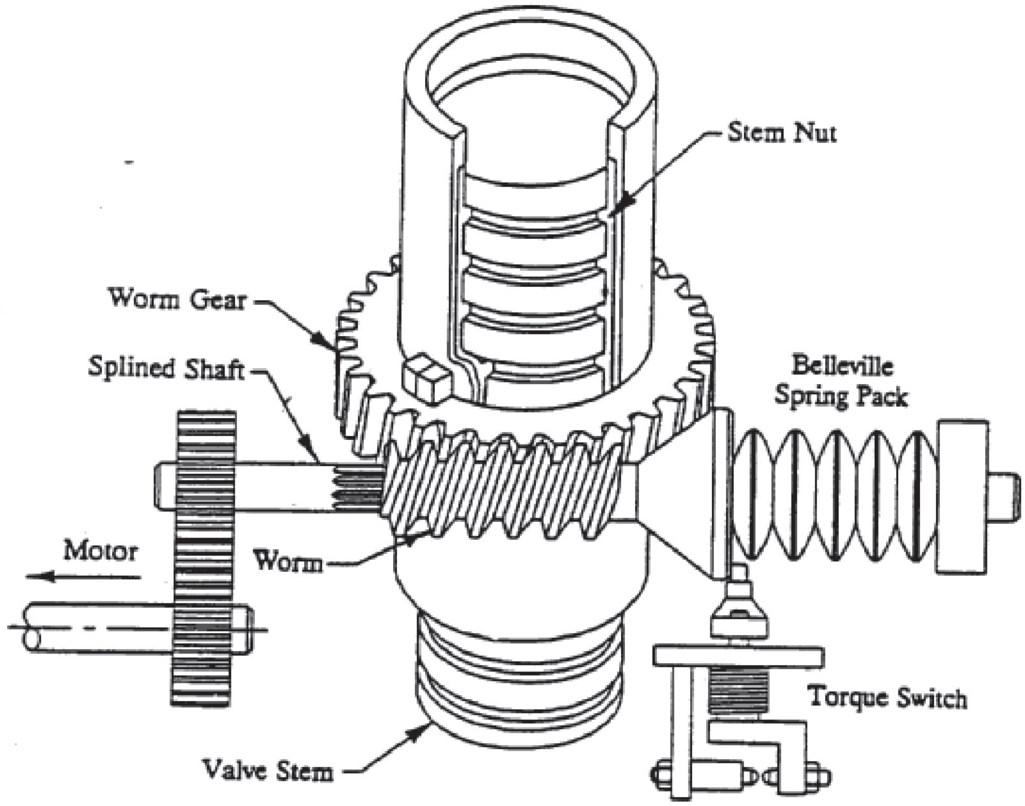

Figure 1 Motor-Actuator Drive Train

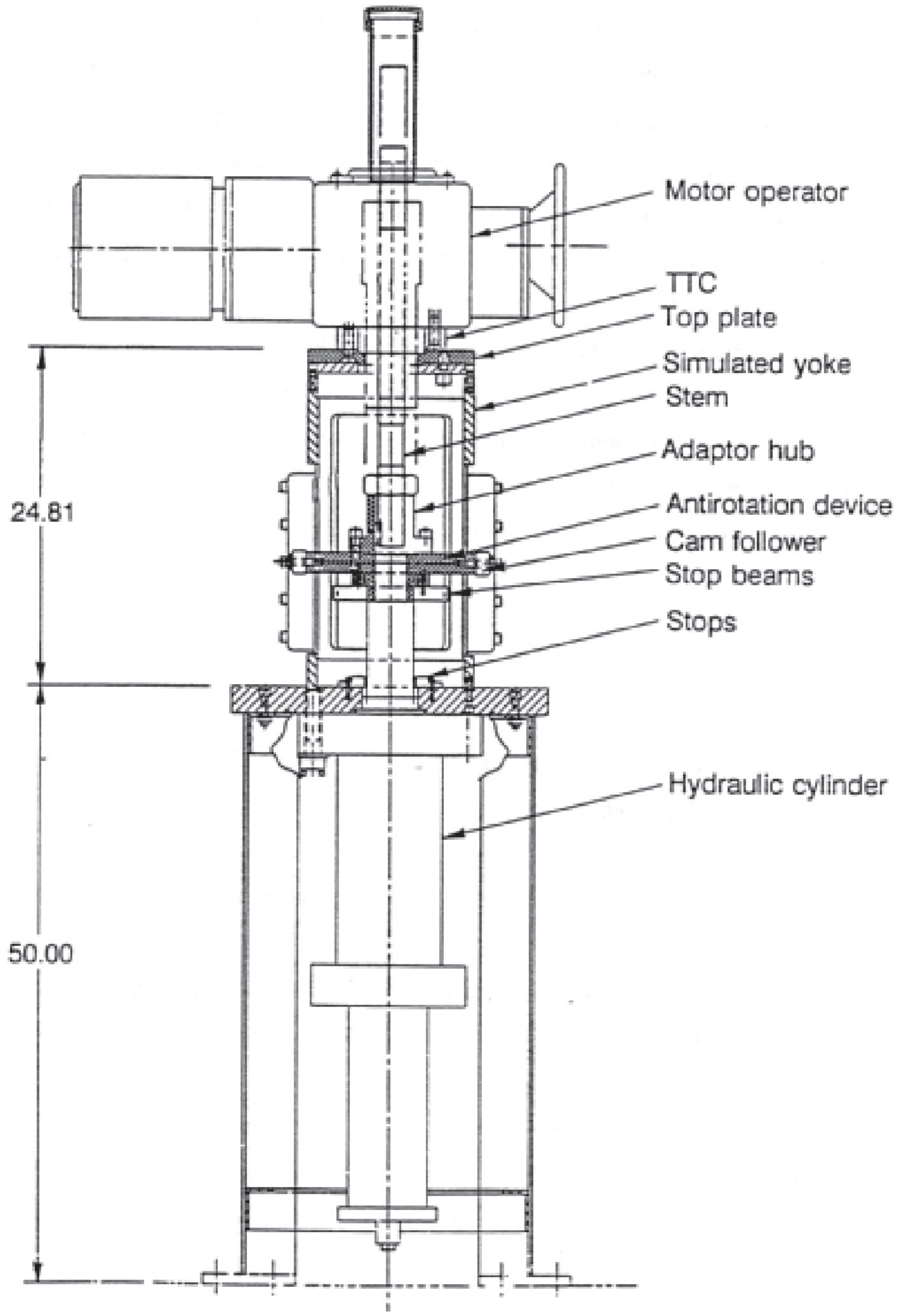

Figure 2 Actuator Test Fixture Components

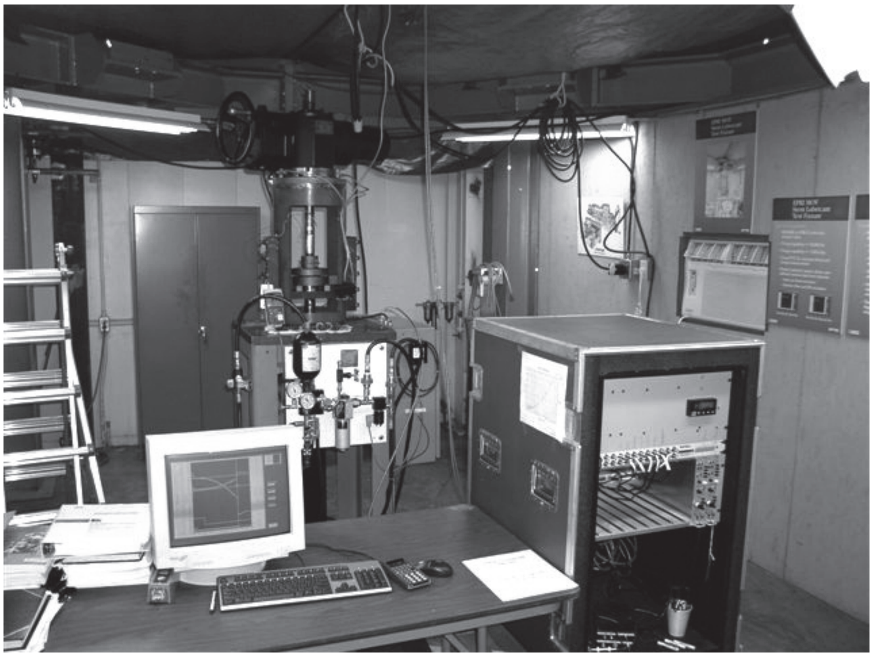

Figure 3 Actuator Test Fixture and Associated Equipment

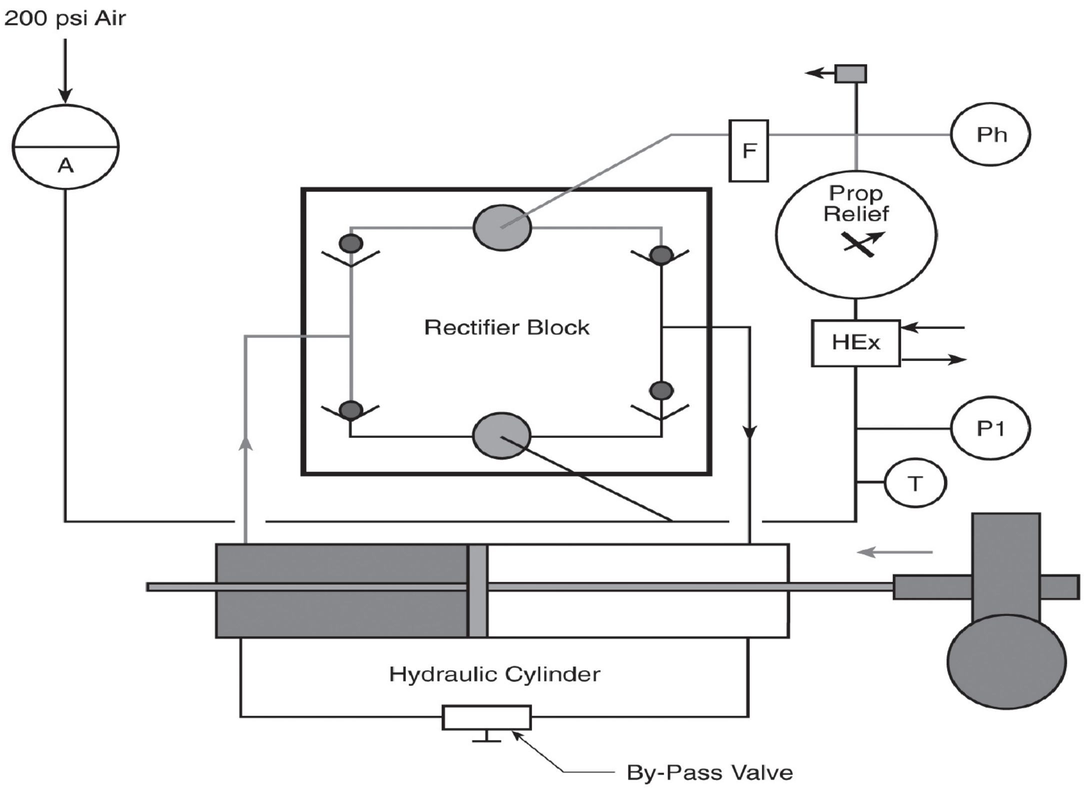

Figure 4 Passive Hydraulic System Simulating Valve Closing Operation

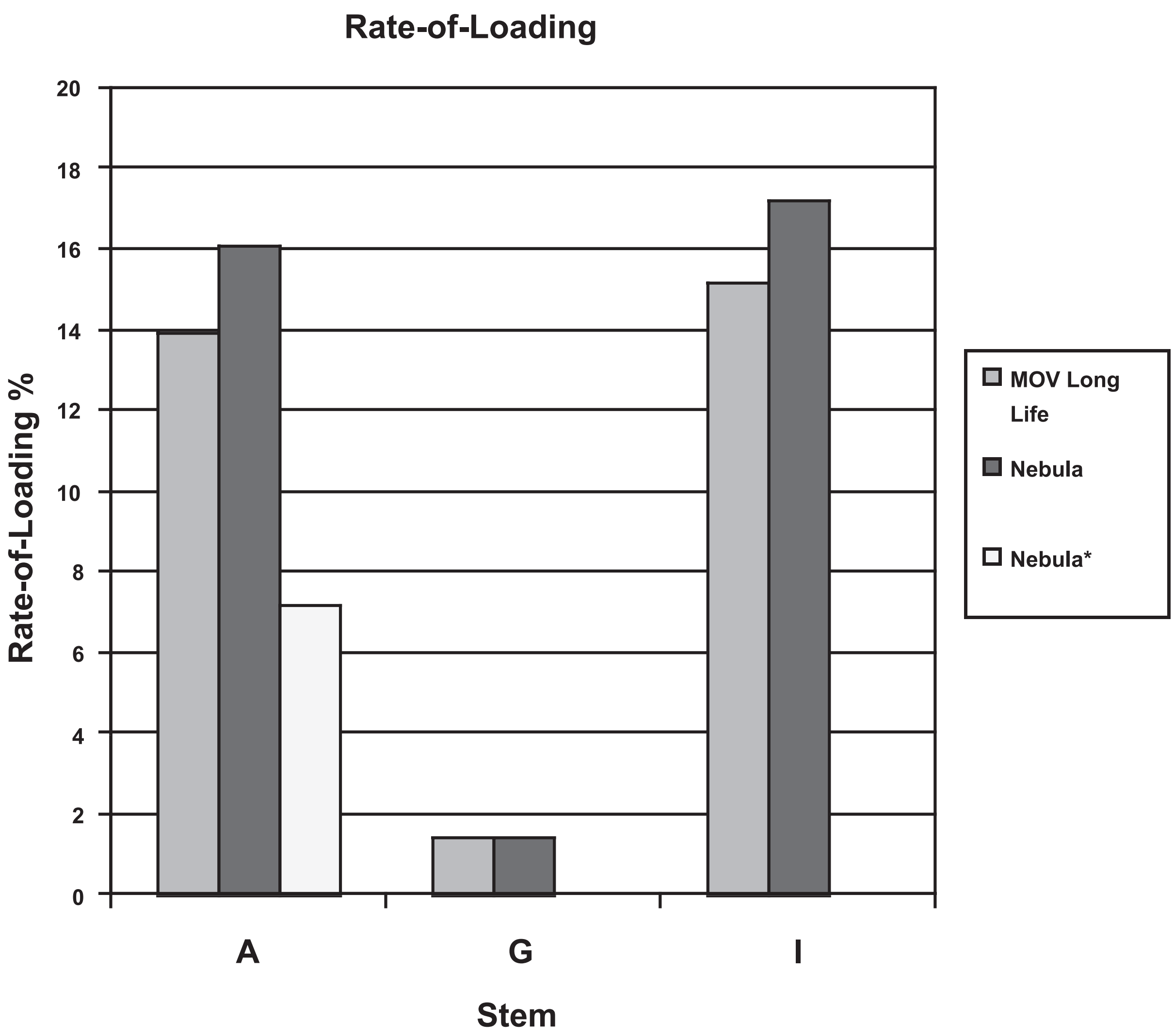

Figure 5 Rate-of-Loading Comparison

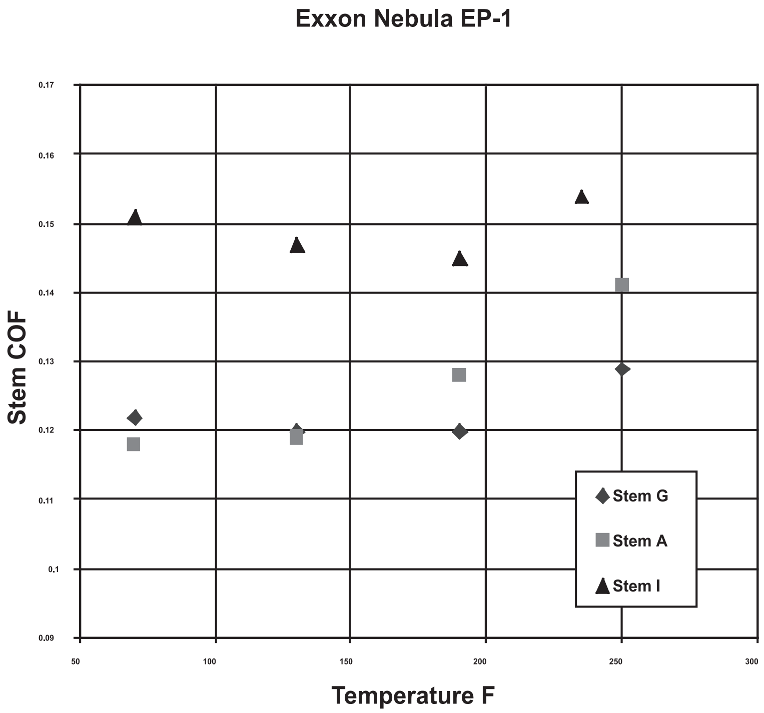

Figure 6 Effect of Temperature on Stem COF – Exxon Nebula EP-1

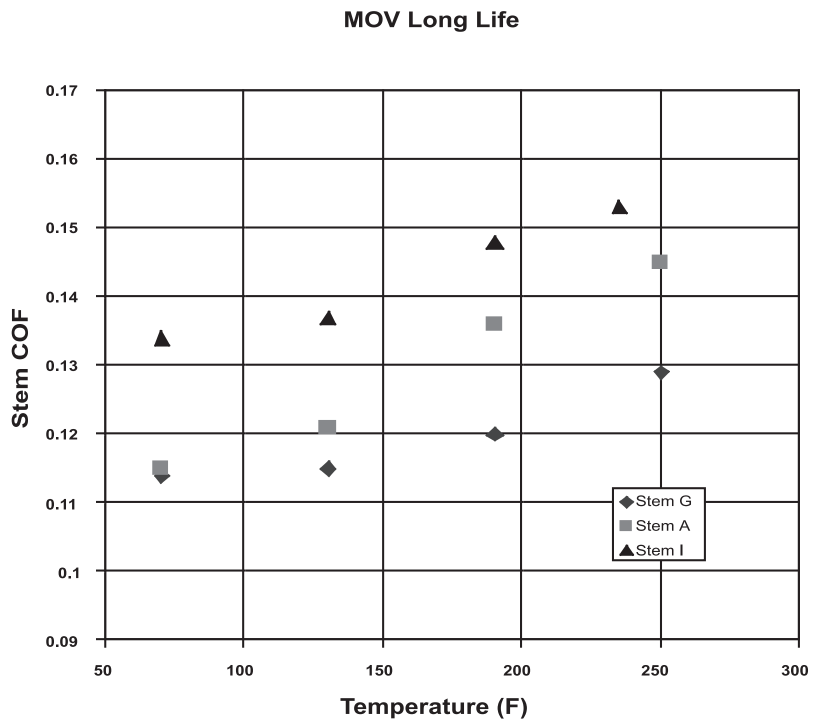

Figure 7 Effect of Temperature on Stem COF – MOV Long Life

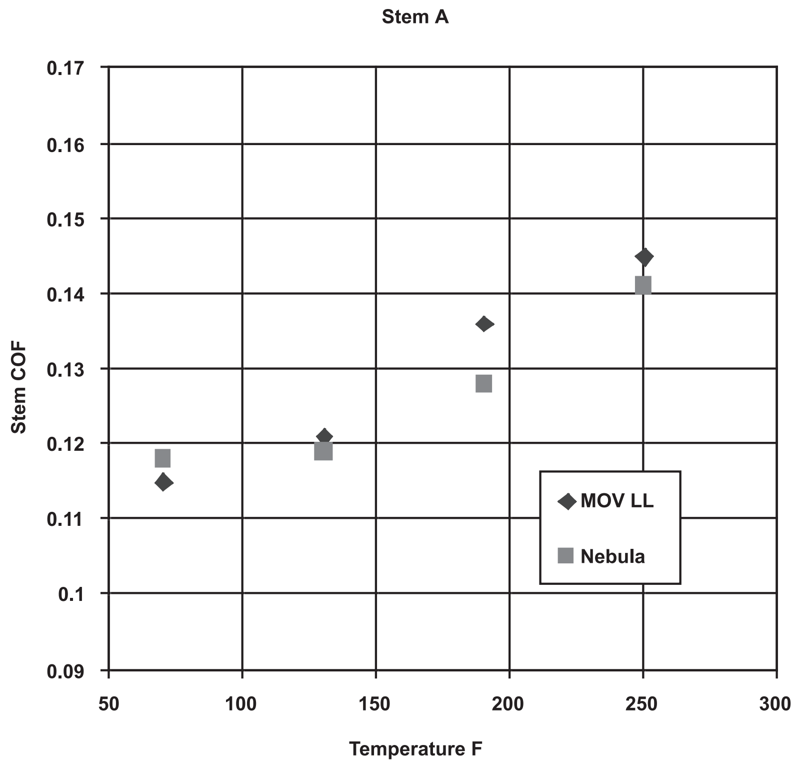

Figure 8 Effect of Temperature on Stem COF – Stem A

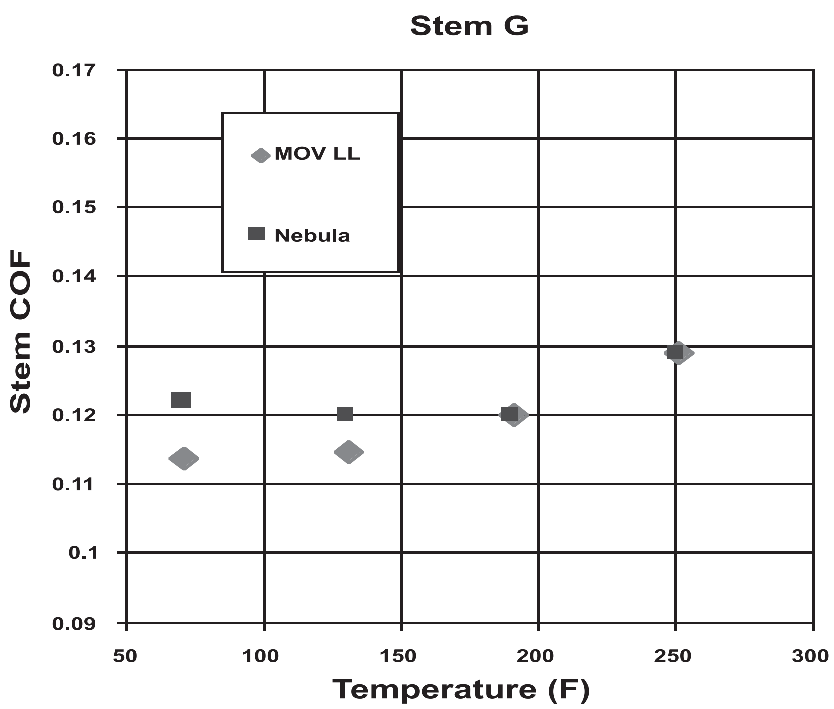

Figure 9 Effect of Temperature on Stem COF – Stem G

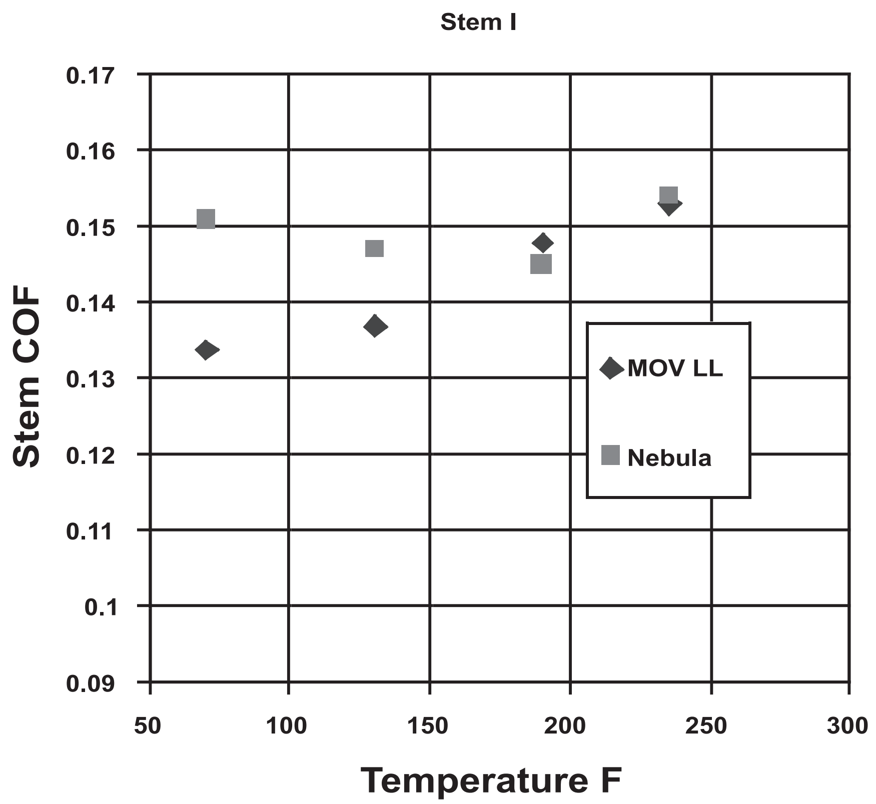

Figure 10 Effect of Temperature on Stem COF – Stem I